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EXPERIMENTAL SPUR-WHEEL REDUCTION GEAR FOR HIGH-SPEED STEAM TURBINES.

CONSTRUCTED FROM THE DESIGNS OF REAR ADMIRAL GEORGE W. MELVILLE AND MR. JOHN H. MACALPINE.

The inherent desirability of interpolating some form of reduction gear between a marine steam turbine and the propeller it drives has long been sufficiently obvious. Suggestions for effecting this by electrical or hydraulic means have been frequent, and, in fact, we hope to illustrate shortly an hydraulic coupling which has given on trial really remarkable results. The straightforward solution of the problem by means of a pinion and a spur wheel has generally been dismissed as impracticable, although the remarkable efficiency of a well-constructed toothed-wheel drive is known to all. The waste of energy in such a transmission gear is notably less than is commercially practicable with an electric coupling, which in turn is superior in this respect to the hydraulic system. Nevertheless, the fact that turbines of the Laval type were not constructed to transmit more than some 300

horsepower constituted *prima facie* evidence that with higher powers special difficulties were to be feared. The remarkable results obtained with motor-car drives, where tooth pressures at times exceed 2,000 pounds per linear inch, were no sure indication that equally good results could be obtained in transmitting by gears some 4,000 horsepower instead of 40, since the rule of three is by no means always applicable in practical constructive engineering. Nevertheless, in his lectures on the steam turbine before the Society of Arts last spring, Mr. Gerald Stoney did suggest that the newer alloy steels and the greater accuracy of workmanship which have made gearing so successful in motor-car practice, did hold out a prospect that the problem of the marine steam turbine might be solved by means of ordinary toothed reduction gearing.

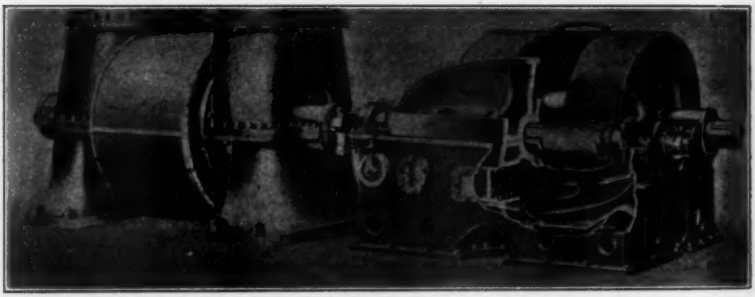


Fig. 1.

Those interested in steam turbines have known for some months past that a solution of the problem on these lines was, with his characteristic enterprize, being put to a practical test by Mr. George Westinghouse, who was having constructed for him a double helical-spur gear intended to transmit 6,000 horsepower at 1,500 revolutions of the pinion per minute, the reduction ratio being about 5 to 1. Through the courtesy of Mr. Westinghouse we are able to publish illustrations of this remarkable experimental gear, the details of which have been worked out by Rear Admiral George W.

Melville and Mr. John H. Macalpine, who have patented the novel features on which they rely for success. A perspective view of the gear, with the casing partly broken away, is represented in Fig. 1, which shows the device coupled to a Westinghouse double-flow turbine. The forgings for the gears were obtained from Messrs. Krupp, of Essen, the teeth being cut by Messrs. Shuchardt & Schütte, Chemnitz, whilst the standing parts were made by the Westinghouse Machine Company.

The pinions have 35 teeth each and the spur wheels 176, a hunting cog being introduced to equalize wear. The pitch is 1¼-inch and the pitch helices are at an angle of 30 degrees with the axis of the shaft. One wheel and pinion have, of course, right-handed helices and the other pair left-handed, so as to eliminate end thrust. The diameter of the pitch circle of the large wheels is about 70 inches and of the pinions 14 inches. Details of the gear box are shown in Figs. 3, 4 and 5. These have been designed after a long and comprehensive study of the circumstances which tend to interfere with the proper working of very broad teeth. A small pitch was deemed essential if a reasonable absence of noise was to be secured, and this necessarily meant broad teeth, in view of the fact that 6,000 horsepower was to be transmitted with a pitch-line speed of very nearly 100 feet per second, and with a limiting pressure of 453 pounds per lineal inch on the teeth.

In order to ensure the proper working of such broad teeth, Messrs. Melville and Macalpine have adopted the plan of mounting the pinion shaft in a device which they have dubbed a "floating frame," the object of which is to cause the alignment and position of this shaft to be controlled wholly by the interaction of the teeth in contact and not by the greater or lesser skill of the workman in laying out and fitting the bearings, which, moreover, even if exactly right to start with, could not be depended on to maintain permanently their alignment. The "floating frame," which is lettered *F* in Figs. 3 to 5, is a heavy steel casting flexibly mounted in the gear box and supporting in rigid bearings the pinion

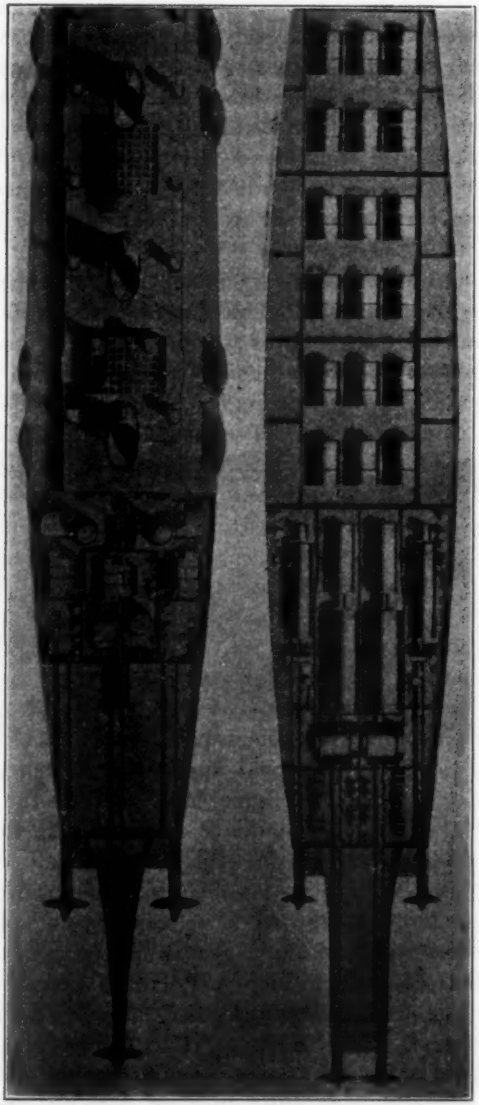
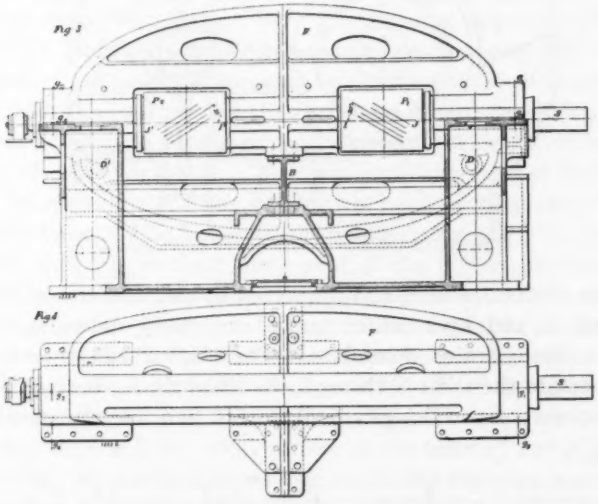


Fig. 2.—Diagram Illustrative of Relative Spaces Occupied by Geared and Direct-Driving Turbines.

shaft, but in such a way as to allow of this shaft having a slight longitudinal freedom, so that it can slide axially to and fro within the frame.

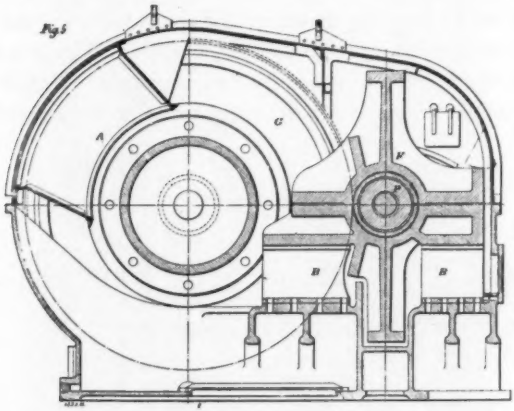
Fig. 3 is an elevation of the floating frame showing the flexible I-beams B, on which it is supported from the bed plate. The pinion, it will be seen, is supported in three bearings, the center bearing dividing the tooth faces P_1 and P_2, of which P_1 is right handed and P_2 left handed. The large gear, being much stiffer, requires only two bearings. The floating frame is made very deep vertically, so as to be exceedingly



stiff to withstand deflection by the nearly vertical forces at the bearings. It is also amply stiffened in the horizontal plane to obviate deflection from the weaker horizontal forces. The floating frame is shown in plan in Fig. 4, whilst Fig. 6 shows the pinion shaft and the flexible shaft S by which it is driven. At the end of this shaft is a coupling C driven by the turbine shaft T. The axes of the gear and pinion are horizontal and parallel to one another when everything is in correct adjustment. But it will be evident that the I-beams B

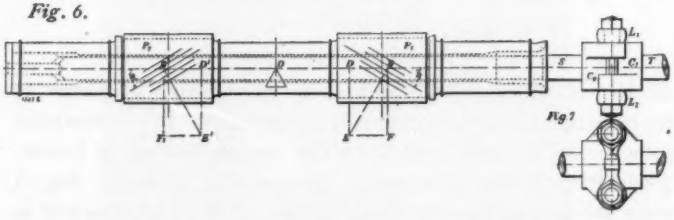
are virtually a hinge, and if the large gear G was not in place, or the coupling C connected, the axis of the floating frame and pinion could easily be deflected in the vertical plane through a small angle by a slight flexure of the webs.

The coupling consists of two flanges, C_1 and C_2, mounted



on the shafts T and S, and connected by two transverse links L_1 and L_2 and by a center pintle, care being taken not to make this restraint redundant. The shaft T can, therefore, only rotate the shaft S through the links L_1, L_2, but as these are transverse no longitudinal forces can be transmitted.

Fig. 6.



Even when seriously out of adjustment the longitudinal forces are so small as to be negligible. The pinion, therefore, has perfect freedom of longitudinal movement in its bearings. Further, the pinion is driven by the shaft S, which passes

completely through it to the end distant from the coupling, where it is keyed and bolted. This shaft is so flexible that it imposes practically no constraint on the pinion and floating frame such as would prevent slight angular yield of the I-beams B. It is evident, therefore, that, both as to its longitudinal position in the floating frame and in the angular position of its axis, the pinion is solely under the control of the forces transmitted by the teeth of the large gear.

Neglecting the very slight friction of the well lubricated teeth, the total forces at the tooth contacts, indicated by B, E and B^1, E^1, Fig. 6, will be nearly at right angles to the teeth at B and B^1; that is, they will be at 30 degrees to the vertical. Hence the two parallelograms of forces shown will be similar. Now the horizontal forces at $B\ D$, $B^1\ D^1$ are the only axial forces acting on the pinion. If these are not equal the pinion will at once shift longitudinally, which it is free to do, till they are equal. The two parallelograms of forces then become equal to one another in every respect, and the vertical force $B\ F$ is equal to the vertical force $B^1\ F^1$. But besides this longitudinal movement of the pinion the frame in which it is mounted is free to rotate about the center point O by flexing the I-beams. Hence it follows that the moment of the vertical force $B\ F$ about O must be equal to that of $B^1\ F^1$ about the same point; and as $B\ F = B^1\ F^1$ the arms of these levers $O\ B$ and $O\ B^1$ must be equal. Thus it is claimed that not only will the total forces on the teeth P_1 and P_2 be equal, but the distribution will be so similar that the centers of pressure B and B^1 will be similarly placed.

This, it is contended, practically assures a good distribution of pressure under all conditions of load. For, by sufficient running of the gear, it may readily be assured for one particular load; and if the pinion and floating frame are made sufficiently stiff this condition will not be seriously changed by increase or decrease of load. The strength of both pinion and floating frame are far in excess of that requisite to sustain the maximum forces to which they will be subject, and their dimensions were made such as to give ample rigidity. For

instance, under full load the flexure of the cast-steel floating frame in the vertical plane is so slight that the end bearings will be lowered relatively to the center by not more than $\frac{1}{2000}$ inch.

There is, moreover, a certain compensation for this deflection, owing to the action of the lubricant. Professor Osborne Reynolds' researches on the action of the lubricating films in Mr. Beauchamp Tower's well known experiments on lubrication seem clearly to indicate that at 1,500 revolutions per minute of pinion in the present gear the oil film in the bearings will be well over $\frac{1}{1000}$ inch thick. When the load on the gear is increased the end bearings will be forced down, bending the pinion and making the teeth bear harder at the inner ends I and I^1. Thus more than the proper force will be exerted on the center bearing, and less on the outer bearings. This will at once lead to a thinning of the oil film on the center bearing, and to its thickening on the outer bearings. As the total thickness of the oil film is more than twice the yield of the outer bearings this change in the thickness of the film will go far to compensate for the elasticity of the floating frame.

Given that the axes of the gears are parallel to start with, the action of the teeth will, it is claimed, force the axis of the pinion into exact alignment with that of the large gear, provided that in the flexure of the I-beams supporting the floating frame the pinion shaft tilts in a vertical plane parallel to that containing the axis of the large gear. If there were no floating frame, and all bearings of the pinion and gear were cast in one bedplate, the longitudinal dimensions remaining unchanged, an error of alignment of the axes in the vertical plane not exceeding $\frac{1}{1000}$ inch in the length over the bearings would entirely upset the uniform distribution of the pressure along the faces of the teeth. Hence, with the most careful work, the above assumptions may not be fulfilled with sufficient accuracy. Even if they were exactly fulfilled when the gear was first completed, inequality of the side wear of the bearings might cause the axes to stand at a sensible angle in the horizontal plane.

Provision is made for measuring such errors of alignment as follows: Outside the end bearings of both pinion and gear, as shown at g_1, g_2, g_3, g_4 in Figs. 3 and 4, there are vertical and horizontal gauges placed so that errors of alignment can be readily measured when the gear is running. If the horizontal gauges at g_3 should show the axes too close by, say, $\frac{1}{10}$ inch, and those at g_4 should show them too open by $\frac{1}{10}$ inch, giving a total error of alignment of $\frac{1}{5}$ inch in the length between the gauges, it may be shown that, with involute teeth, which are here used, they will bear hard at I and I^1, Fig. 3, and lightly at J and J^1. If the teeth were perfectly rigid there would, of course, be point contacts at I and I^1, and the teeth would actually stand open by a small amount at J and J^1. This opening Messrs. Melville and Macalpine call the "opening of contact."

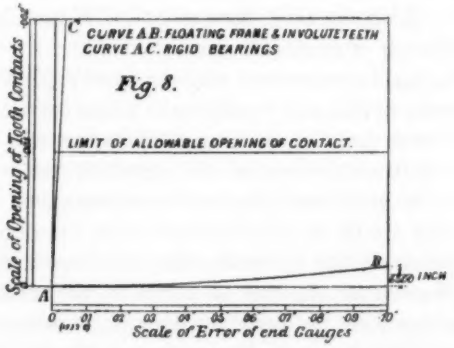
If in changing from perfect alignment the pinion turned about either a vertical or a horizontal transverse axis, it may readily be shown that the opening of contact would be a considerable fraction of the sum of the errors shown by the end gauges. On the other hand, the inventors state that the action of the floating frame in combination with involute teeth is such as to constrain the pinion to turn about an axis parallel to what is known as the line of action of the teeth—in the present case, an axis inclined 14 degrees 30 minutes to the vertical—and this is precisely the movement which reduces the opening of contact to a minimum.

The result is shown in Fig. 8. The error of each end gauge is indicated by the horizontal scale, the total length representing $\frac{1}{10}$ inch, corresponding to a total error of alignment in the horizontal plane of $\frac{1}{5}$ inch between the gauges, as explained above. The vertical scale shows the "opening of contact" if the teeth are rigid, this scale being enormously magnified, as the quantities to be represented are so small.

The curve $A\ B$ represents the resultant opening of contact for given gauge errors for the present gear. The extreme opening of contact shown is $\frac{1}{6500}$ inch, and this for a gauge error which should never be approached. At half this gauge

error, still large and easily measurable, the opening of contact is reduced to one-quarter of the foregoing, or $\frac{1}{25000}$ inch, a quantity far below the limit of accuracy of the very finest machining.

Consider next the effects of the elastic strains of the teeth and shafting. Steel gears of $1\frac{1}{4}$ inches pitch are commonly run under a load of 1,000 pounds per inch of tooth face at ordinary pitch-line speeds. As all gears are liable to errors of alignment this average pressure actually means that at times there will be a maximum pressure very much greater. In the gear under discussion the highest mean value used is less than 500 pounds per inch of tooth contact. Suppose,



then, that by errors of alignment the actual value varied from zero to 1,000 pounds per inch, which is far within the disturbance occurring in ordinary gears. It may be shown that the yield of these teeth under 1,000 pounds per inch is such that the distribution of pressure would close up an opening of contact of nearly $\frac{1}{1000}$ inch. Thus, even under the extreme gauge errors shown at *B*, Fig. 8, Messrs. Melville and Macalpine claim that the disturbance of distribution of contact pressure is comparatively slight, and at half the gauge error stated it would be insensible. They claim, therefore, that gauge errors of sensible magnitude do not produce a sensible disturbance. Errors of erection will be exceedingly minute,

so that the only errors to be avoided are those due to unequal horizontal wear of the bearings. Any unequal vertical wear will be compensated for by slight tilt of the floating frame. As the forces on the floating frame are inclined to the vertical at only a small angle, vertical wear will take place much more quickly than horizontal; and as the inequality of the wear will usually be but a small fraction of the total wear, it would be almost impossible for any very sensible angular error in the horizontal alignment, such as assumed above, to creep in before the bearings were worn down a long way vertically, and ready for renewal. The case they state is very different if the floating frame is dispensed with, and all the bearings are cast in one bedplate. The worst supposition is that there is an error of alignment of the axes in the vertical plane. Keeping all dimensions of the gear and bearings the same as in the gear already built, the resulting "opening of contact" is, they find, shown by curve *A C*, Fig. 8. This exhibits extreme sensitiveness to this error of alignment, as allowable limits will be passed before each vertical gauge shows an error of $\frac{1}{1000}$ inch. Such an error would be so small as to be quite insensible to the eye, and would arise even from strongly screwing down one of the holding-down bolts, or from slight heating of the bearings, and could not be avoided by the most perfect workmanship. Hence the gear with rigid bearings they hold to be quite impracticable.

Involute teeth have been used, as they give the excellent results shown by the curve *A B*, Fig. 8. But they have also the well known property that if, through wear of the bearings or errors in the original setting up, the axes are somewhat farther apart than designed, the teeth still run true, and there is no "opening of contact." This does not hold with any other form of teeth. For instance, with epicycloidal teeth the action will not be true if the centers move apart, the axes remaining parallel; and in these fine high-speed gears a recession of the centers not exceeding $\frac{1}{1000}$ inch would, the designers assert, be quite serious. With this form of teeth also, when the alignment is defective, an opening of contact

of $\frac{1}{1000}$ inch, supposing the teeth rigid, corresponds to an error of each end gauge of $\frac{1}{100}$ inch—that is, a total error of alignment between the gauges of $\frac{1}{50}$ inch, which could hardly be detected. Besides, on account of the way in which openings of contact occur with epicycloidal teeth, the limit of allowable opening of contact is, it is claimed, much under $\frac{1}{1000}$ inch. Hence this form of tooth is far inferior to the involute on account of its extreme sensitiveness to errors of alignment, and also from the fact that it will not run properly when the axes part even slightly, so that the stresses would be much intensified, and the action of the teeth would be noisy. The curve *A B* (Fig. 8) is tangential to the base-line at *A*, while for all other arrangements and forms of teeth the curve of opening of contact cuts this base-line at an angle. Hence the solution presented by the floating frame with involute teeth is unique.

From the foregoing it will be seen that the elasticity of the I-beams allows freedom of movement of the axis of the pinion and floating frame in the vertical plane. But by warping the webs of these beams the axis of the floating frame might also turn slightly in the horizontal plane. It is, of course, many times more rigidly mounted against this displacement than against that in the vertical plane, but still the rigidity is by no means complete. Now it may appear that this is not specially important, as from curve *AB*, Fig. 8, it would seem that such an angular displacement in the horizontal plane, even if of measurable amount, causes no important disturbance. But that is not the only consideration, as the question then arose as to whether with freedom of rotation, both in the vertical and horizontal planes, the position of the axis of the floating frame would be stable or unstable; that is to say, if forcibly displaced from exact alignment to a small angle in the horizontal plane, will the forces at the tooth contacts tend to bring it back to perfect alignment or to further displace it? By consideration of the slight changes of direction of the forces at the tooth contacts produced by such displacement, it can readily be shown that, with involute teeth, the

position of the axis of the floating frame is unstable, and a slight displacement arising from any cause will tend to increase. With the I-beams alone, therefore, there might arise excessive horizontal errors of alignment, and trifling causes might produce sudden and violent changes of the horizontal error, either of which effects would endanger the whole mechanism. The floating frame has, therefore, been deprived of all freedom of motion in the horizontal plane by means of two horizontal struts standing transversely between the floating frame and bedplate. These bear on the floating frame at D and D^1, Fig. 3, and one is clearly shown in Fig. 1. Obviously these struts in no way interfere with the freedom of movement of the floating frame in the vertical plane. Where they bear in the bedplate there is an adjusting and locking mechanism which greatly facilitates the true setting up of the floating frame, and is to be seen in the perspective view in the front of the bedplate, near the end next the turbine.

With epicycloidal teeth in perfect adjustment, the position of the floating-frame axis can be shown to be stable without struts. But this is only an apparent advantage, as this stability would at once be upset by widening the gear and pinion centers or by the slightest wear. But even if this stability could be trusted, it is believed that epicycloidal teeth are far too sensitive to errors of adjustment to be used, and the question of stability or instability in no way affects this consideration.

The only condition which could cause serious disturbance of the tooth pressures is an excessive heating of the pinion above the temperature of the gear. But the design provides for a copious application of lubricating and cooling oil, especially to the pinion, which has most tendency to heat. The cover, also, is so arranged as to draw in air at the ends and at A, Fig. 5, by the fan action of the gears, and discharge it by openings (not shown) to the right of this cross section. Besides, water can be circulated, by means only partly shown in the figures, between the pinion and the flexible shaft S. But it is not anticipated that the heat to be removed per

minute will be so great as has been provided for. The gears of heavy electric trains work under much higher stress, both in the metal and between the teeth in contact—very much higher, as these stresses are accentuated by errors of alignment shown by rounding of the teeth through wear—and with practically no lubrication. Yet they run for long distances. In the reduction gear these unfavorable conditions are reversed—there is good lubrication and very uniform contact. If the coefficient of friction at the tooth contacts is $\frac{1}{10}$, it can readily be shown that the frictional loss at the teeth will be under 1 per cent. So that even if the gear were transmitting 6,000 horsepower—the very highest power hoped for—this frictional loss would be under 60 horsepower. As at least one-half will go to the large gear, where it will readily be dissipated from the large surface, there is left well under 30 horsepower to remove from the pinion which, with the various means provided, should present no difficulty. If, then, this control of the heat is assured, there will, it is claimed, be practically perfect tooth contacts; this, with the fine pitch and the spiral gears (avoiding sudden entering or leaving contact of one whole tooth at a time) will, it is asserted, ensure reasonably quiet running of the gears.

At 1,500 revolutions per minute of pinion and 453 pounds per inch of tooth contact, the gear will transmit 6,000 horsepower. The pitch-line speed would then be 5,500 feet per minute and the mean speed of sliding about one-tenth this, or 550 feet per minute. There is on record a case of a right-angled worm gear run successfully at 15,000 feet per minute with a contact pressure of 350 pounds per inch. The average speed of sliding will here be greater than the pitch-line speed by over 41 per cent., or, say, 21,000 feet per minute. Taking speed of sliding multiplied by pressure as a rough means of comparing these two cases, we get:

$$\text{Reduction gear, } \ldots \frac{5{,}500 \times 453}{\text{Right-angled worm gear, } 21{,}000 \times 350} = \frac{1}{29.5}.$$

The extreme constants chosen, therefore, seem well within practical limits.

Before the experiment is tried it is impossible to predict the limit of safe load, but the following are two cases which it will be interesting to consider:

1. At 1,000 revolutions per minute of pinion and 283 pounds per inch of tooth contact—that is, as shown above, only 28 per cent. of the intensity of pressure used with steel gears of 1¼-inch pitch at ordinary pitch-line speeds—the gear would transmit 2,500 horsepower. The famous merchant ship *Aberdeen*, by the engines of which Dr. Alexander C. Kirk introduced triple expansion, was about of this power. Her main engines, without shafting, propeller, stern tube, spare gear and fittings, weighed 221 tons. If we deduct from this the condensers, pipes, sea valves, air pumps, circulating pumps, donkey pumps, floors and gratings, etc., the weight left for the main engines proper will be about 150 to 160 tons, say 150 tons. Hence the following comparison may be made:

	Tons.
Westinghouse marine turbine of 3,000 brake horsepower, 1,000 revolutions per minute, weighs . .	27
Reduction gear, ...	25
Total, ...	52
Aberdeen's engines replaced, ...	150
Saving, ...	98

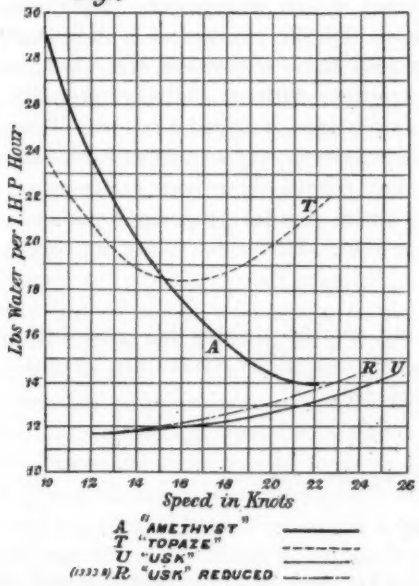
That is, 65 per cent. of the weight is saved. If this design will transmit 2,500 horsepower, it is claimed that similar gears could be applied with great saving of weight and space to all steamships up to 18 knots, which comprise over 95 per cent. of all the ships afloat.

2. As already stated, at 1,500 revolutions per minute of pinion and 453 pounds per inch of tooth contact (or only 45 per cent. of the intensity of pressure for similar teeth at ordinary pitch-line speeds), the gear will transmit 6,000 horse-

power. If this be realized similar gears could be applied to any speed of ship. This design would then be of the correct power for the *Dreadnought* and, applied to her, would save fully 50 per cent. of the weight of her turbines, besides reducing boiler weights, since both turbines and propellers would now be of considerably enhanced efficiency.

Increasing, by the law of comparison for similar machines, the dimensions and power of the present design from 6,000

Fig. 9



horsepower, a size is obtained suitable for the *Mauretania* with three large screws of the same total power as the present four-screw ship. Here, again, the weight of the turbines would, it is claimed, be halved, as also the engine-room length. The boilers, also, as in the *Dreadnought*, would, it is considered, be materially reduced.

In Fig. 2 the upper view represents the *Mauretania* thus modified, as compared with the existing arrangement shown

in the lower of the two views. For warships the saving effected by such a gear should, it is claimed, be considerable, especially at cruising speeds. In Fig. 9 is reproduced a curve *A*, showing the coal consumption of the *Amethyst* as compared with the *Topaz* (curve *T*). The latter, however, was not fitted with reciprocating engines of maximum efficiency, much better results being obtainable where the conditions are better suited to the reciprocating engine. The curve marked *U* shows the consumption of the *Usk* on the assumption that at full power the evaporation was 8 pounds of water per pound of coal, and at 13 knots 9 pounds. The curve lettered *R* is the *Usk's* curve reduced to a smaller scale of knots, so as to make the full speed the same as that of the *Amethyst*. On these assumptions it will be seen that the

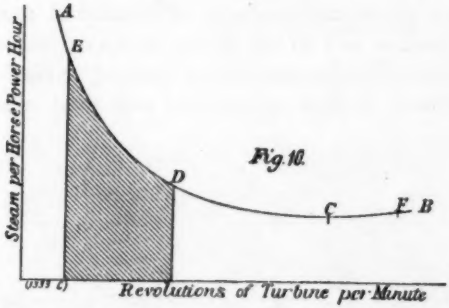


Fig 10.

reciprocating engine had the advantage, but, in justice to the turbine, it must be added that much better results have been obtained with the newer turbine cruisers. Nevertheless, the fact remains that marine turbines are commonly underspeeded.

In Fig. 10 is plotted a curve showing the relation between steam consumption and the revolutions of the turbine. The portion of the curve frequently used in marine practice is hatched, and shows clearly how very rapidly the efficiency falls off as the speed diminishes. By using gearing it is contended that the turbine at maximum power can be even slightly overspeeded, as at *F*, so that at cruising speeds the efficiency will actually be increased.—"Engineering."

THE NORMAND BOILERS OF THE SCOUT CRUISER *CHESTER*.

By Lieut. A. F. H. Yates, U. S. N.

The *Chester* is equipped with twelve Normand water-tube boilers installed in three watertight compartments, four in each compartment. The general arrangement is shown in figure I. They are designed for a working pressure of 250 pounds and were tested to stand 400 pounds. The safety valves are set at 268 pounds. The ratio of grate surface to heating surface is 1 to 46, being 58 square feet of grate surface against 2,670 square feet of heating surface. The total grate surface is 696 square feet and total heating surface

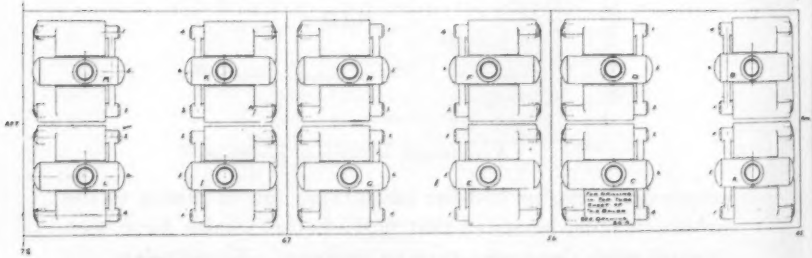


Fig. I.—Arrangement of Boilers, U. S. S. "Chester."

32,040. The length of grates is 7 feet $2\frac{1}{8}$ inches and their width 8 feet $\frac{7}{8}$ inch. The boilers each have one furnace with three furnace doors. The height of the boilers is 12 feet $5\frac{5}{8}$ inches, length 11 feet $6\frac{5}{8}$ inches and width 11 feet 4 inches. They each contain two nests of tubes numbering in all 986. The tubes are of the cold-drawn seamless-steel type, No. 10 B. W. G., $1\frac{3}{8}$ inches outside diameter, and of lengths varying for each row. There are four smokestacks arranged as shown

in figure I. The forward and after stacks each take from two boilers and the two middle stacks each take from four. Their mean height is 63 feet 3 inches. The area of each of the smaller stacks is 17.72 square feet, and of the larger stacks 35.78 square feet. Forced draft is provided for by the closed-fireroom system. Forty-nine per cent. of air space is allowed for in the grates.

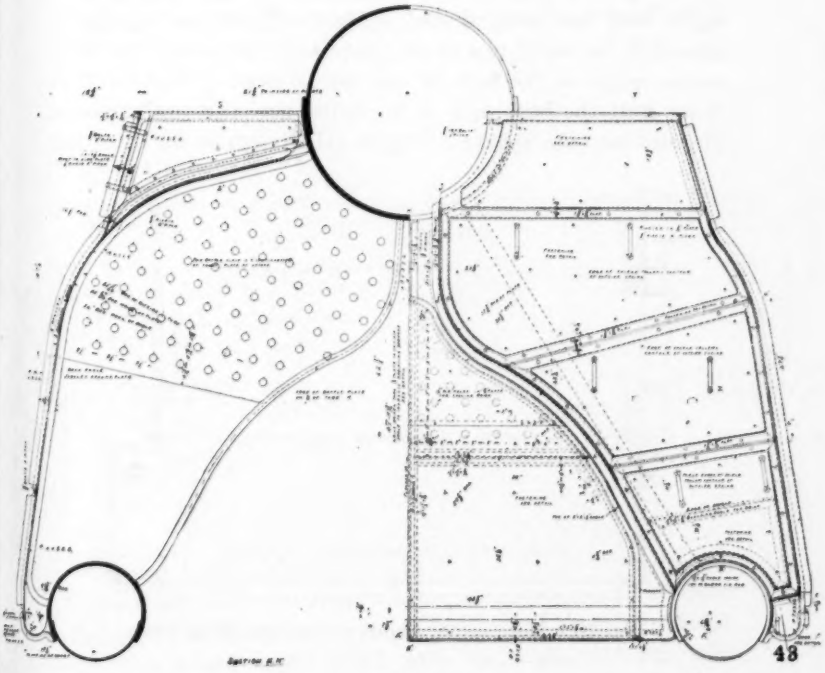


Fig. II.

SECTION. BACK END.

Figures II, V and VIII show the elevations of the boiler, which is composed of a large steam drum with steam dome attached, two smaller water drums at the bottom and two down-comers connecting steam with water drums, also a set of generating tubes on each side. It is braced by tie rods

and by the down-comers. The down-comers stay the steam drum to the water drums in the front, and diagonal hollow tie rods stay it at the back. Front and rear horizontal tie rods secure the water drums to each other at front and back, respectively.

The tubes are of the bent type, of a more pronounced bend even than figure II indicates. The inner rows on either side for the greater part of their lengths from the back form a tight wall by being spaced closely. Tubes are similarly spaced in the outer row of each side and form a wall for their entire length at the bottom, but are so bent at the top that those near the back end of the boiler are open to the escape of gases into the uptake. Figure III illustrates the direction

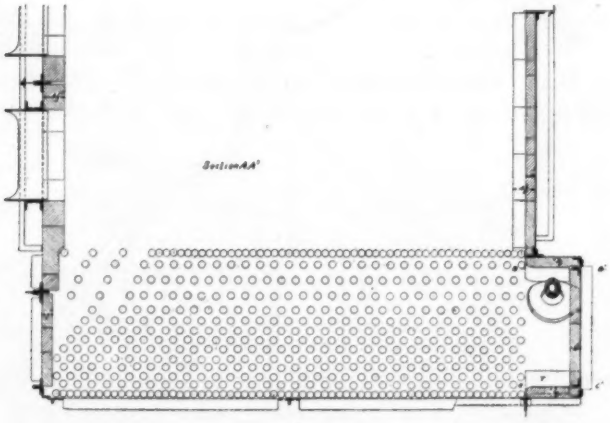


Fig. III.—HORIZONTAL SECTION THROUGH TUBE NEST.

taken by the gases during the process of combustion. Tubes are staggered in the intervening rows of tubes. All tube ends enter the drums normally. The smoke stacks are at the back ends of the boilers, directly over the uptake. The uptakes have a clear area of about one-seventh that of the grates.

The boiler casings are made in sections designed to be air-tight. Figure VIII shows the front of the boiler as encased.

At the sides there is an inner casing, a sheet of asbestos board, a 2-inch space filled with magnesia, and an outer casing. At the front and back of the boiler there is a wall of fire brick 2 inches thick opposite the furnaces in the upper portion and about 4½ inches thick for a height of about one foot above the grate. The sides of the furnaces are also lined with fire brick for a height of about one foot above the grate, protecting the inner row of tubes where their ends enter the water drums. The 2-inch wall at the front end extends behind the tube clusters out to the side casing. At the front and back of the

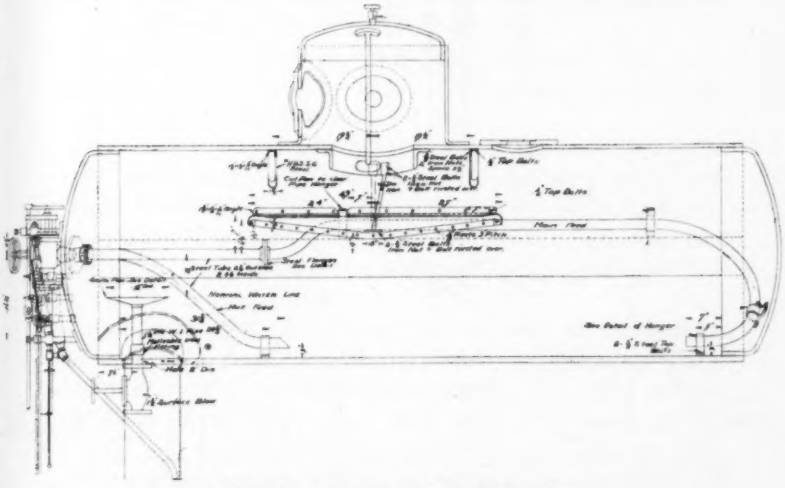


Fig. IV.—Section of Steam Drum.

boiler there is also an inner casing, an air space and an outer casing. The boiler front has an additional covering consisting of a 1-inch space filled with magnesia, and an outer casing. Doors are fitted in the casing front and back where necessary for the cleaning of soot from the tubes. Air holes are cut in the lining and in some of the fire brick to keep the brick and casing cool.

The drums are all made of two plates with longitudinal seams lapped and riveted, and with drum heads flanged and

riveted on their ends. Manhole plates are fitted where necessary. The steam drum is 40 inches internal diameter and the water drums each 18 inches. The connecting down-comers have an internal diameter of 11¾ inches. Figure IV shows the construction of the steam drum and its dome.

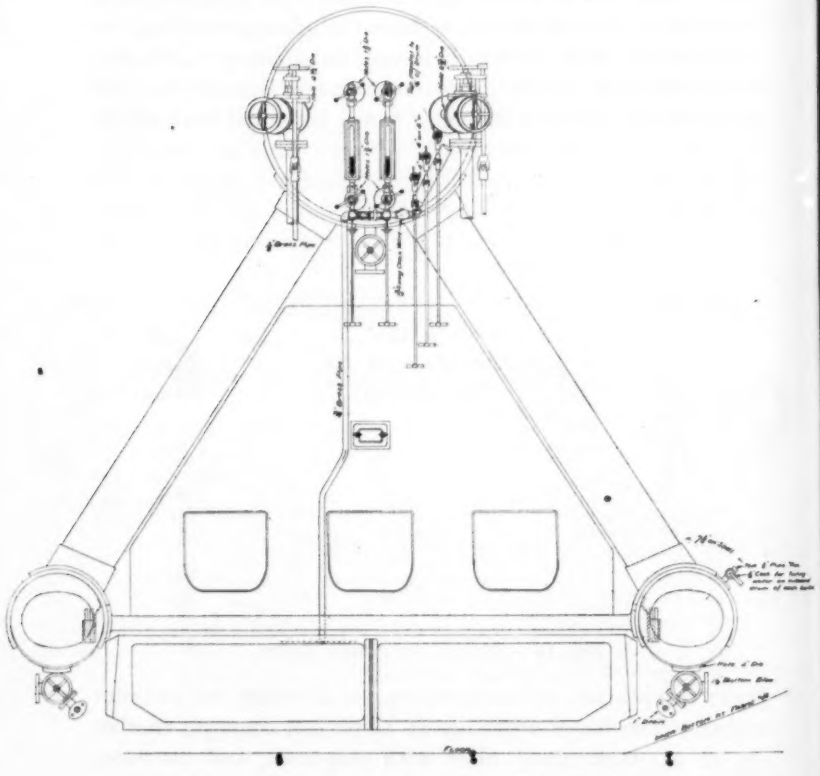


Fig. V.—Position of Fittings on Boiler Front

The circulation of water is very rapid. Steam generated in the tubes rises to the steam drum and starts a circulation which continues downwards through the down-comers to the water drums and upwards again through the tubes. The

diagonal tie rods which secure the steam drums to the water drums at the back, being hollow, assist in the circulation.

The furnace doors are fitted to swing inboard and to be held open by a latch. The ash-pit doors swing in a like manner on a ratchet. A mica peep hole is fitted through the front casing of each boiler for the purpose of observing the fires.

Soot boxes are constructed on each side of the boiler along the bottom of the outer rows of tubes to collect the soot which works through the tubes at this point. They are best shown in Figure II. Small hinge doors are built at each end of them through which to blow out the soot.

The grate bars are of the built-up type, of rolled steel, with lightening holes to reduce their weight. They are open to allow for 49 per cent. air supply.

Over each furnace there is a 1¼-inch pipe connection with spray nozzle for putting out the fires by spraying water over them. These pipes take water from the fire main, and provision is made for operating the system either from the fire-room or the deck above. An additional connection with the sea has also been installed.

Immediately over each boiler, in the deck of the blower room overhead, there is provision made for operating main and auxiliary stop valves, fire extinguishers and safety valves.

The tubes are swept twice daily when underway, during the mid and afternoon watches, and but once a day in port, during the mid watch. Steam or air may be used, but the latter proves most satisfactory.

A provision for air supply to the fires is made through the sill which lies between the bottom of the furnace doors and the top of the ash-pit doors. Air entering this point passes up through the boiler casing and into the furnace just inside of the lower edge of the furnace-door frames. Figure VII shows this.

Each boiler has two water gauges, and one glass of the Klinger type and one of the Scotch type are used. A main feed-check valve is fitted on one side of the front of the steam

drum, and an auxiliary feed-check valve on the opposite side. Stop valves are fitted between the check and the boiler shell in each case. The steam stop valves are fitted to the steam dome, the main stop on the front and the auxiliary stop on the side. Three test-cocks are fitted on the front end of the steam drum, and one is fitted on the top of each water drum. An air cock is fitted in the top of the steam dome. Bottom-blow valves are fitted to the water drums, and a surface-blow valve to the steam drum. Figures IV and V show these fittings.

The bottom and surface-blow valves are connected with the overboard-discharge valves in the same compartment by a $2\frac{1}{2}$-pipe, and a pipe of the same diameter connects the bottom-

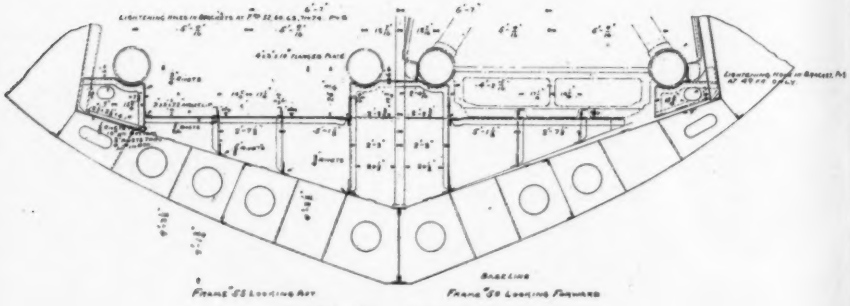


Fig. VI.—BOILER SADDLES.

blow valves with the suction manifold of the fireroom feed pump. Drain cocks are fitted at the bottom-blow valve connection on each water drum.

The internal feed pipes are as shown in Figure II. The main pipe extends throughout the entire length of the steam drum and the auxiliary pipe but half way.

Each boiler has a twin-spring safety valve with two valves, each of $3\frac{1}{2}$-inch diameter. These valves are located on the top of the steam drum, in the rear of the steam dome, where shown in Figure IV.

Baskets containing rolled-zinc slabs are fitted along the inner sides of the water drums. None are fitted in the steam drum.

The boilers were constructed in the shops and lowered into the ship whole. Owing to the limited space allowed for them there was barely room for this to be done, and it was necessary to swing them very skilfully. As installed they are too snug, and the cleaning and painting of a considerable part of their external surface is extremely difficult and, to some extent, impossible. For the same reason it is very unhandy sweeping tubes through the doors in the back of the boiler.

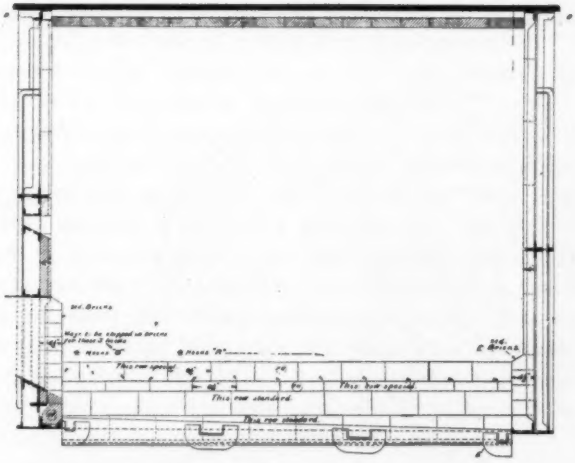


Fig. VII.—VERTICAL SECTION THROUGH FURNACE.

The boilers have foundations as shown in Figure VI. The saddles are placed at proper intervals along the length of the water drums and these latter rest on them.

Very few repairs have been necessary though the boilers have had unusually severe service during the sixteen months they have been in use. Renewing soot boxes, furnace fronts and internal feed pipes is all that has been found to be beyond the capacity of the ship's force. The soot boxes were originally constructed too lightly and with too little volume, which resulted in their burning out. New soot boxes of

slightly different design and of large size were built and no further difficulty has been experienced. The furnace fronts are also of light construction and these burned out, but it was largely due to the brick wall failing. With more experience in making secure brick walls the furnace fronts are standing up well. The renewal of the internal feed pipes was necessary by reason of their corroded condition, and when new ones were installed they were better secured than formerly to resist vibration, and the length and curve of the main internal feed pipes were altered. One alteration has also been made in the form of an additional connection from the fire-extin-

Fig. VIII.—FRONT END, INSIDE.

guisher piping to the sea. No trouble has been experienced in the boiler proper from corrosion. In fact, they are remarkably free from it. Whenever the boilers are opened they show up clean and in generally excellent condition. Success in this regard is attributed to the fact that practically no cylinder oil is used in the department and also to the extreme care taken in fitting the boiler zincs to a perfect metallic contact. A mixture of kerosene and graphite is used almost entirely on the auxiliary engines. A point worthy of mention, perhaps, is the fact that no boiler zincs are fitted in the

steam drum. It is true that the internal feed pipes have corroded badly, but it is after rather long service, and no corrosion is evident on the shell of the drum. Likewise the ends of the tubes are but very slightly corroded. The boilers are, as a rule, simply cleaned with fresh water. No kerosene has been used for the past year and soda has only been used on two occasions, about twenty pounds being used each of those times. As regards the boiler zincs, it has only been necessary to renew them once to date in those boilers that have had the greatest steaming service, and but twice in the others.

Except in seams in the water drums no leaks have been experienced, and these have only been of minor extent. They are caused by the construction of the water drums and the way in which they rest on the boiler saddles.

Figure VI, while not in detail, will give an idea as to the way they rest on the saddles. The longitudinal seams of their lap-riveted joints are very close to the boiler saddles, and they undergo a strain for this reason. The saddles are placed at intervals under the water drums, and where the latter rest on them the leaks are most likely to occur, especially at the end, where the drum-head is flanged to the drum proper. Many parts of these seams are inaccessible owing to the too snug installation of the boilers. None have been experienced worthy of note since the time new soot boxes were installed, when an opportunity was offered for calking them.

No tubes have had to be renewed, nor have any ever evidenced leaks. Five per cent. of each size and shape of tubes are carried in case necessity arises for renewal. It is thought, however, that it would serve better if these spare tubes were straight and of the length of the longest size. They could easily be bent to template and cut to exact length. They would offer greater resource and be easier to stow.

Though the working pressure has usually been about 260 pounds the Klinger gauge glasses used have been most satisfactory. Only a fair number blow out, and when they have done so it has always been from a reasonable cause. At first many were broken because the men fitting them did not

understand how much care should be exercised, but later the only cause had been corrosion due to use and time employed. It has been the practice to use graphite in fitting them.

A point which has been noticed when using the surface blow is the discoloration of the water in the gauge glass when the water is dirtier than usual. It is due to the location of the scum pan, the latter being too close to the gauge-glass connections. So little trouble is had, however, from dirty water in the boilers that no change is considered necessary.

It might also be mentioned that the type of gear fitted for operating the test cocks is such that occasional difficulty is experienced from coal dust clogging the threaded stems which operate these cocks. It is of no great importance, however, and is an evil difficult to overcome.

Some few improvements have been found possible in the brick work which extends along the water drums at the sides of the furnace. A special form of brick has been devised which does away with fourteen special sizes formerly employed. Then, again, vertical pins are used in securing the bricks in lieu of S-shaped dogs originally fitted. The firemen frequently ruined portions of brick work by striking these dogs with their firing tools.

Mention was made above of a provision made for admitting air through a space in the sill below the furnace doors on into the furnace at the top of the fires. This space was so prominent that the firemen would scatter coal dust in it when lifting their shovels to feed the fires, and a hood has been built over the opening to screen it.

It is noticed that the finish work of shoving back the coals and keeping the fires in shape at their front end when tending them would be greatly facilitated if the furnace doors were lowered and their lower sill on the same level as the grates. Being slightly elevated this operation is slightly awkward. Another point is the effect of deadening the fires at the front end when the ash-pit doors are thrown open, due to shutting off the air supply. Means are being considered for swinging these doors open with a dropping motion so as to overcome this.

But little use is found for the mica peep-hole in the front of the boiler casing. It is difficult to clean and almost impossible to keep so. There would appear to be little need for its existence.

Cast-iron grate bars were supplied the ship in small numbers on an occasion of emergency, and they have proven quite satisfactory, having outlived others of the built-up steel type. They were too heavy for the bearer-bar supports to stand up under, so that the latter were reinforced. They are less expensive and if carefully cast should prove of greater worth.

The difficulties experienced are on the whole of little moment and are few and far between, but the boiler is in some ways delicate and must be given every attention. The boilers on this vessel, however, have stood as severe a test as service conditions could be expected to demand, even in war time, and it must be accepted that they are an excellent type. They have stood two four-hour and one twenty-four-hour official maximum full-power trials, furnishing steam for between 18,000 and 24,000 brake horsepower, and in addition have stood a large number of protracted official service-endurance trials. The total distance over which they have been used since being installed is in excess of 36,000 miles, and perhaps some idea may be had from knowing that what amounts to five sets of firing tools have been worn out during this service.

Dry steam is always produced, and priming is a very rare occurrence. The boilers stand forcing to the extreme limit before priming, and it is only when the demands are most unreasonable that it results from this cause. It is easily caused by the water being salty, however, even if it is to the slightest degree so.

Two hours are usually taken to raise steam and connect the boilers from clean grates, though on one occasion in an emergency two boilers were prepared and put in use eighteen minutes from the time the order was given. One good example of the efficiency of the boilers is the case of a standardization trial when speeds were run from ten to twenty-six

knots, all boilers being in use and the speeds being up and down. Thirty-three runs were made over the measured mile in less than eight hours and the boilers stood for a successful run each time.

The usual method for running is under $\frac{1}{2}$-inch air pressure, as natural-draft conditions prove unreliable ordinarily. The time-firing device is always used and a three-shovel standard followed. The thickness of the fires carried varies from four to ten inches, dependent upon the air pressure. On no occasion has more than $2\frac{3}{4}$-inch air pressure been averaged, though the blowers have sufficient capacity for greater pressure.

In most ways the boilers may now be considered in even better condition than when first installed, though perhaps more extended service is necessary to fully tell the tale.

THE DAVISON REVOLUTION INDICATOR AND REGULATOR.

By Lieut. W. B. Tardy, U. S. N.

An instrument to indicate at a glance the actual number of revolutions made by the engine is a most essential aid to keeping exact distance in squadron and to adjusting the engine quickly to the exact number of revolutions during maneuvers and to keeping a certain definite constant speed of engine whenever this is specially required, as when standardizing over a measured course, going across the range at target practice, and when observations of change of range are to be made.

Numerous instruments for indicating revolutions have been tried with more or less success. Among those that have been installed in various vessels may be mentioned The Monitor Speed Recorder and The Kaiser Chrono-Tachometer, which has been installed on a number of vessels with very satisfactory results. The following is a description of another instrument of this character invented by Mr. G. C. Davison, of the Electric Boat Company.—H. C. D.

The device consists essentially of the following parts: Refer to Figure 2; the vertical shaft S″, carrying a worm wheel at its lower end and a miter wheel at its upper end; this small worm wheel T is connected by means of a worm S′, shaft and bevel gear to some part of the main engine in such a manner that shaft S″ makes one revolution for each revolution of main engine. On this vessel the worm and worm shaft S′ are connected to the vertical shaft of the main-engine counter. The miter gear on upper part of vertical shaft S″ meshes with another miter gear on horizontal shaft S‴; this

shaft in turn carries a hard bronze friction wheel, or pinion F, and is a little larger in diameter than the miter gear on shaft S'''. The position of friction pinion F on shaft S''' is determined by setting of the micrometer screw E and micrometer wheel O. F is made to revolve with shaft S''' by means of a feather and key. The micrometer arm E is graduated by fives from forty-five revolutions to one hundred and twenty revolutions, and the micrometer wheel O is graduated by tenths from zero revolution to five revolutions, so that the instrument is capable of being set for revolutions from forty-five to one hundred and twenty, by tenths.

The locking nut D holds the setting for revolutions in permanent place. There is an arm B which carries on its inner end a small roller whose object is to hold the friction disk K' in constant frictional contact with the friction pinion F. The friction disk K is supported top and bottom by ball bearings, and carries with it a velvet-lined pad H, on which rests a timepiece, in this case the timepiece is a large, ordinary commercial stop watch. The gear connections to the engine, or moving part driven from the engine, are such as to cause the timepiece to be carried in a counter-clockwise direction. The whole mechanism is enclosed in a brass casing and looks very much like a small compass binnacle. It is mounted on the floor plate at the working platform in the starboard engine room so that the throttle machinist looks directly down on the timepiece and graduated circle in Figure 1.

Referring to Figure 1, the circle on the top of the instrument, which is fixed, as far as the mechanism is concerned, is graduated from zero to one hundred and eighty degrees, in either direction. This graduation is worked out for this particular instrument from the distance (5.6 yards) actually made by the *Vermont* on her builders' trial for every revolution of her propeller. Necessarily this distance and, therefore, the graduation on this top rim will change with different conditions of draught and foulness of bottom, but the inventor has furnished curves of correction to be applied for varying conditions.

FIG I

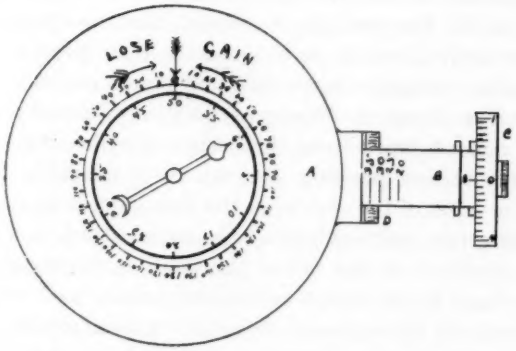
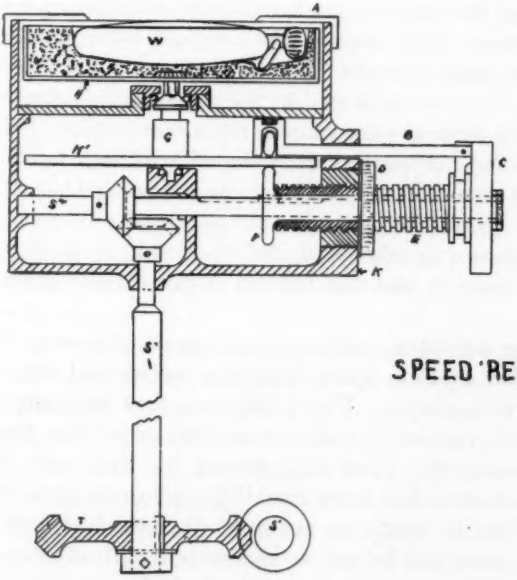


FIG II



SPEED REGULATOR.

The operation of the revolution indicator is as follows: The timepiece being wound and fitted in its place on the velvet-lined pad the hand will make one revolution every minute. By means of the micrometer wheel and screw the instrument is set for the revolutions ordered, then, as long as the throttle is manipulated so as to maintain this speed, the hand of the watch virtually stands still and points constantly to the radial arrow shown in Figure 1. If at any instant the engine room receives word from the bridge that the ship is, say, forty yards behind position, the cap A is shifted with reference to the hand of the watch, in the direction indicated by the circular arrow, marked "gain," the radial arrow would then be forty divisions to the left of the hand of the watch; and since the hand of the watch moves at constant and uniform speed it would be necessary to impart a faster counter-clockwise motion to the timepiece than it previously had, in order to bring the hand of the watch into coincidence with the radial arrow. *Vice versa*, if the ship had been ahead of position by a given amount the top rim would have been shifted in the direction indicated by the curved arrow, marked "lose," and, in this case, the counter-clockwise motion of the instrument would necessarily have to be reduced by throttling down until the hand of the timepiece caught up again to the radial arrow. It is the work of only an instant to set the instrument for any given speed, and to lock it in place by means of locknut D, and the top rim A can also be shifted instantly.

This device has been used at every opportunity since its installation and uniform speed has been maintained with a great degree of accuracy. There has been very little opportunity to test it out in formation, since this vessel has been in fleet formation only three days during the time that the revolution indicator has been installed, and even then the leading ship was so erratic in her speed that the instrument could not be tested out for any considerable length of time.

When the device was first installed all of the throttle machinists were averse to its use, regarding it as an additional

complication; but after a very short experience every chief-machinist's mate who stands throttle watch became perfectly familiar with its operation and uses it voluntarily at all times, even to the extent of resetting it when the speed is changed temporarily, to half speed, for instance. If for any reason the throttle machinist is absent momentarily from the throttle, and the watch hand and arrow separate in either direction, all he has to do is to gradually increase or decrease the speed of the engine until the hand and radial arrow are again coincident, and he knows that he has made the average revolutions for the entire time.

Experience seems to indicate that the graduation on the micrometer screw and wheel are not quite correct. This is to be expected, since the instrument was never experimentally calibrated, but this is of minor importance. For instance, if the speed is sixty-four revolutions and the engine room is notified that the ship is forty yards behind position, we shift top rim A, and gain the forty yards; if, after the lapse of another interval of time, the ship is again behind position, we know that the instrument is not set for the correct speed, assuming that the leader is maintaining a constant speed, and we therefore change the setting of the micrometer from 64 to 64.1 or 64.2, and after one or two trials the instrument is correctly set for the speed that it is desired to make.

It is thought that the only weakness in the instrument is the remote chance of the failure of the friction, but this seems very remote.

If every ship of the fleet were equipped with one of these revolution indicators and regulators, properly calibrated and graduated, distances would be better preserved and with considerably less attention on the part of the throttle machinist.

The natural and essential corollary of revolution indicator and regulator of this character is some visual means of signalling revolutions from the bridge to the engine room, and of indicating from bridge to engine room the amount ahead or astern that the ship is off her position. The following has been suggested: An electric transmitter on bridge, with re-

ceiver at gauge board in starboard engine room, to another receiver on the bridge, all in series, by means of which the officer of the deck may signal ahead or astern, 1, 2, 3, . . . 9, the zero (0) in each case being assumed. Thus, with twelve wires it would be possible to transmit signals of position from ten to ninety yards in either direction; also some step-up arrangement installed in some manner as above, for indicating the prescribed revolutions, from thirty, say, to one hundred and twenty, so that the officer of the deck makes contact for sixty-four revolutions, for instance, and if his receiver on the bridge shows sixty-four he will know immediately that the identical signal is displayed in front of the machinist at the throttle; this display to continue until the revolutions are changed and some other display is exhibited.

THE CHARACTERISTICS OF COPPER UNDER VARIOUS CONDITIONS.*

By Mr. Geo. W. Handscomb.

Much has been written within recent years upon copper and its peculiar characteristics, for the most part from the metallurgical and statistical side. Little appears to have been put forward from the practical tradesman's point of view, and this paper is an attempt to deal with the manipulating of this somewhat expensive metal at the hands of the copper worker. Attention will be given more particularly to the working of copper for engineering purposes, both marine and land requirements, the domestic and other purposes to which the metal is put not concerning us for the moment.

The whole process of copper working is a closed door to many engineers not brought into contact with the coppersmith's shop, and it is more especially to such that my effort is addressed. Possibly the more widely-experienced engineer will be able to gather some useful and interesting data from my remarks. Each process will be explained in detail, and in such a manner that the comparative stranger to the actual working of copper will, I hope, after reading this paper, be enabled at any stage in the process of pipe making to determine fairly accurately the quality of material and workmanship before him. In this manner I hope to restore, or gain, the confidence of the users of copper pipes. The notion that so-called trade secrets should be exclusive is altogether antiquated. Publicity leads to better understanding all round, and the superintendent engineer or the overseer is always more satisfied when familiar with the process of manufactur-

* Read at the Institute of Marine Engineers on 22d March, 1909. Reprinted from "The Steamship."

ing the commodity he is called upon to inspect. Were this more generally recognized, we coppersmiths would have less occasion to deplore the substitution of other metals. It is to be regretted that many otherwise good tradesmen work by "rule of thumb," which is, of course, no rule at all. This is peculiarly mystifying to the beginner, and accounts for the many diverse methods of working in different shops. The great majority of coppersmiths have little or no conception whatever of the reasons for adopting a certain method. Having seen it done before, they are quite satisfied and merely follow old customs. The scientific engineer will at once perceive the weakness of this method, and will, no doubt, recognize in the remarks I shall make that no development will be described unless the object aimed at or reasons for its adoption are stated.

Deposited Copper.—Much reliance is placed in some quarters upon various mechanical tests for determining the quality of the copper, usually, I fear, misplaced. This craze for test pieces received a severe shock at the time of the Elmore panic. In this instance we had a metal capable of passing all the recognized mechanical tests twice over, yet this metal, after mere exposure to the weather for a year or so, without any attempt to put work upon it beyond that which the tube and sheet people had executed, literally fell to pieces upon efforts being made to work it. Without in any way submitting a chemical or scientific explanation for this phenomena I would take the liberty of enlarging upon a theory of my own, which it may repay us to follow up; at any rate I offer it for what it may be worth. It is that the molecules or particles of copper, the cohesive force of which give the metal its peculiar ductile characteristic, being mechanically mixed, that is, deposited crystal upon crystal in the electric bath, offers much less resistance to the action of atmospheric gases than does the metal mixed in the molten state. In the latter procedure, the poling of the copper by the insertion of a sapling, well stirred into the molten metal, introduces chemicals in the form of gases, which act as a protection against ex-

ternal atmospheric attack. When copper is heated to brazing heat, or an even less temperature for a long period, a transformation appears to take place in the copper, this peculiar protective chemical formation, due to the poling, escapes, and the metal becomes brittle and useless for copper piping. Without going so far as to accept the somewhat extravagant hypothesis of the micro-organic theory in regard to metals, I am firmly convinced, by close observation, that there must be certain atmospheric gases which have a decidedly deleterious effect upon copper: call it a disease if you wish, but it is beyond dispute that degeneration takes place in copper under the action of gases. This is quite apart from common oxidizing or rusting. Moreover, in regard especially to deposited copper, this weakness can be demonstrated to be due not to the amount of work put upon the metal, but solely to the length of time the metal is exposed to the atmosphere. Whatever the weakness is, its development is abnormally rapid, as within the short period of a few months a piece of this Elmore copper, capable at first of withstanding extraordinary strains, becomes absolutely useless. This deposited copper could be gradually worked with repeated annealings (sometimes annealed over one hundred times) into a class of special work, such as expansion pieces of various designs, and it would finish with a surface absolutely clear of marks or defects. No other copper ever equaled it in this respect, yet if, when finished (at which stage it would be remarkably ductile), it were exposed to the atmosphere for from two to three years, or if under steam pressure with the corresponding heat for a few months, the result was disastrous, and the copper gave every appearance of breaking into small pieces. I have dealt at some length with the now somewhat ancient topic of Elmore copper for the simple reason that I believe the comparison of experience in this, compared with other metals, will give us the clue to some of the hitherto unexplainable difficulties. I am anxious that readers will bear in mind my digression and deductions, which will occasionally be referred to later.

Mechanical Tests.—Our study of deposited copper naturally leads us on to the subject of mechanical tests. I fear that the doubling over and bending of copper strips into various more or less fantastic shapes is of little final value. As a matter of fact, mechanical experiments, unless spread over a long period of time, are useless for authentic data purposes. The orthodox Board of Trade and Lloyd's test for doubling over an annealed strip of copper and thinning down the doubled edge with a hammer is unsatisfactory. As I have never, in over twenty years' experience, found a well prepared test strip fail, yet I have certainly at times had some very poor copper through my hands, but this test fails to detect the weakness. If the craving for a mechanical test must be satisfied I would suggest that a test for detecting any latent weakness in seamless tubing be as follows:

Carefully smooth-file round the end of a piece of the tube to be tested, and after annealing, work over a flange with a hammer, the width of flange to be, say, $\frac{3}{4}$ inch for 2-inch bore tubes and above, and $\frac{1}{2}$ inch wide for below 2 inches, the radius at neck to be twice the thickness of the material. This should be clear of any signs of splitting or breaking. It is a fairly stiff test, and would undoubtedly show up any tendency the tubes may have for splitting longitudinally, and also be a fair test for transverse weakness.

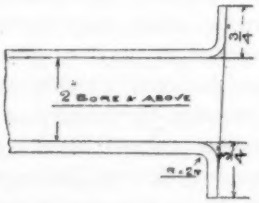


Fig. 1.—FLANGING TEST FOR SEAMLESS TUBES.

My own personal experience, however, in regard to this mechanical-test question has been unsatisfactory, good copper sometimes giving bad test results and poor fracture, at another time copper showing good results in the testing and first-rate

fracture has in succeeding stages developed abnormal weaknesses, even after the most careful personal supervision in regard to temperature, etc. What I suggest is that the buyer should take note of the most important details in regard to pipe manufacture, and from what is shown in this paper judge for himself if he is being treated honestly, and in that way gain confidence in the capacity of the coppersmith to detect any faults and to replace same with good material. Where care has been taken to procure tubes and sheets from reputable makers comparatively little trouble follows.

Corrosion.—The question of corrosion or wasting is one that comes more under the jurisdiction of the chemist or analyst than the practical coppersmith. The coppersmith can but note results and theorize. I will give a few very striking instances of the effect of this wasting in copper piping. It will be worth the engineer noting the effect upon copper steam pipes of the patent covering sometimes used, composed of dung and other mysterious ingredients, the ammonia and other products from the covering acting upon the copper and causing pitting. The fact that the copper is pitted, and not acted upon uniformly, may be explained in two ways: first, the particular solvents may not be of equal strength and activity over all parts; secondly, the metal may not be completely homogeneous; as a matter of fact no metal is absolutely so. The extraordinary effect upon copper steam pipes on locomotives, when fixed in the way of the furnace gases, is remarkable. A method adopted upon some old-fashioned outside-cylinder locomotives of fixing the copper steam pipes in the smoke box at front end of the engine occurs to my mind. These pipes became sadly pitted and scored where the hot gases played upon them, and I have seen copper pipes $\frac{3}{16}$ inch thick under these conditions wasted completely through. Strange to say, in the case of a seamed pipe under these conditions, the solder was not affected and stood out upon a wasted pipe sometimes like a piece of cord along the pipe. Heat undoubtedly aggravates the tendency to corrode when the particular chemicals and gases are present. I would sug-

gest, if a durable and suitable paint can be procured, that all pipes be coated over, not necessarily to hide the coppersmith's bad work, but to exclude the atmospheric and other gases.

Purity of Metal.—There seems to exist a somewhat erroneous idea that copper to be really trustworthy and durable must possess a small amount of impurities. No one appears to have any conclusive idea as to the exact quantity of this foreign ingredient necessary. The matter is usually mentioned in a delightfully hazy manner that the impurities must not be too much, nor must they be too little. I have not the slightest difficulty in understanding the smelter's position in regard to this impurity question. The smelter's greatest trouble is to produce an approximately pure copper, and he, for obvious reasons, is not enamored with the idea of producing an absolutely pure metal. In fact, by the ordinary smelting process a pure copper cannot be produced. The problem of how engineers and many coppersmiths come to regard impurity in copper as a *sine qua non* is not difficult to solve. It is the result of attempting to determine the virtues of copper by the same methods applied to steel and other metals, that is by tensile and elasticity tests, and the various mechanical tests mentioned previously. It does not follow in the case of copper that the piece giving the highest tensile figure or showing the greatest elasticity limit is the best for the purposes of pipe making. Copper pipes rarely give way because the pulling stress between the points at which it is fixed is too great, but as a result of a series of vibratory movements gradually hardening the material; this I shall deal with in the section upon "overstraining." Some copper with a fair share of impurities in it gives splendid test figures, but completely fails when put under any prolonged working, either by the coppersmith or in its final position under pressure and vibration. The best metal I have ever had through my hands, giving the best results both in its production and final utility, was copper electrically pure, but smelted up again and cast into billets and drawn or rolled. I cannot see any ultimate solution to the question of purity in copper until the metal-

lurgist and the peactical worker are more in touch with each other. Meanwhile I pin my faith, for the reason given, to the pure copper.

Overstraining.—A most important point to be noted in the working of copper, either tube or sheet, and probably this applies equally to many other metals, is that although in the process of annealing after hardening by working, the metal is returned to its original ductile condition, this has its limits. What I wish to infer by that is that copper can be overworked or strained beyond the point at which annealing will return it to its previous malleable state. This is exactly what happens to pipes when they are kept at work unduly without being annealed; not only are they liable to failure at any time whilst in that strained condition, but if annealed they will have restored to them their original quality only in a limited degree. Engineers naturally are suspicious that it is not so much the preservation of the copper pipe the coppersmith is anxious about, as the placing of another item upon his order book and later upon his ledger account. To such I can only say, read carefully this paper and then draw your own conclusions. In regard to this overstraining by the copper worker I have had exceptional opportunities of observing this in the many highly-worked pieces of copper passing through my hands. When one has to laboriously hammer by hand pieces of copper for days sometimes, to find at the end that the copper has failed, it makes one concentrate his faculties for seeking out causes, and the failure is often due to overstraining in trying to work the copper excessively to facilitate progress. It is obviously difficult to give definite advice on this particular subject; to say that copper should not be worked too much, neither too little, is somewhat nebulous; experience must be the final arbiter. Questions of gauge in the material and shape of work enter so much into the problem that fixed rules cannot be given, but if the data given in this paper for bending pipes and working bends be not exceeded all overstraining will be avoided. Overstraining in the working of copper shows itself by irregular cracks,

which after being cut out with a gouge chisel reappear. When any of these cracks appear it is not only a sign of overstraining, but also a sign that the piece of copper showing the cracks should be treated by the brass founder in his melting pot, in a word "scrapped." These longish irregular cracks that develop must not be confused with the patches of irregular marks that sometimes show, and which give one the impression that if knocked up from the inside the piece would fall out; this is due to an entirely different cause, and will be touched upon in the section upon annealing.

Annealing.—The process of annealing gives, I dare venture to say, by far the most concern to the outsider. Any defect in a pipe, of whatever description, is at once put down to faulty annealing, that is, overheating. I am able to write with some amount of authority upon this subject, having carried on experiments and noted results covering a long period of time. It is of little use to a genuine inquirer to have made to him the bald statement that a particular defect is due to burning or overheating, unless some authentic reasoning is produced to support the assertion. It is useless, as a rule, making complaints to the manufacturers of the tube or sheet in respect to any defects coming to light in the working of the copper; they have one stereotyped reply to all comers, which is that the copper has been overheated. The statement is the same, even if the copper has never been near the fire since its receipt from the mills. This burning notion is a fairly safe line to take, unless the person dealing with the copper has had a long experience and been observant. In similar respects to the question of impurities in copper previously mentioned, this subject of the temperature at which copper loses its characteristics is equally vague and arbitrary. The difficulty arises through efforts being made to trace all defects back to overheating, hence confusion and contradiction. I shall try to classify the trouble.

The actual temperature to which the copper is heated is not nearly so important as the length of time it remains at the given heat. I am not giving the temperatures of the

different colored heats, for the simple reason that an instrument for its detection is not usually part of the equipment of a coppersmith's shop, and the coppersmith must depend upon a much more rough-and-ready method for daily use. Now I should define burning as raising a piece of copper to such a heat (as a matter of fact, white heat) that caused the copper to become roughened and covered more or less with globules or blisters. This condition could not be mistaken, though it might be better to take a piece of copper one's self and try it. Strange to say, if the piece of copper has been treated in this manner, presuming the operation to have been rapidly performed and the copper only slightly roughened, it apparently has received little damage and will stand all the ordinary tests that would be applied to the piece not so heated. This conclusion is not arrived at by hasty methods, and it amounts to this, that heating copper to any temperature short of blistering the surface has little ill effect upon the copper, though all the same, I do not advocate trying how near one can get without actually blistering the surface, because it is not necessary to raise copper to even brazing heat to properly anneal it. What is a real danger in the heating of copper is the prolonged heating. The terms "gassing" and "reducing flame" are new names for an old trouble, with which intelligent coppersmiths have long been familiar; many may not have known the actual chemical process, but the evil was recognized. Copper tube or sheet can safely be raised to a bright red heat without danger; the burning heat is considerably higher and, unless with gross carelessnes, is rarely ever reached, but the sheet or tube must be immediately removed upon attaining to the heat mentioned. Any discussion upon the rival merits of quenching in water or natural cooling would be "much ado about nothing." Cooling off in water shows a distinct disadvantage in the case of a very intricate and prolonged piece of work requiring some dozens of anneals. This I have proved time and again in the same manner, and for the same reasons as in the case of overstrain. For ordinary work, however, requiring but one or two an-

nealings, the difference is so trifling as to be negligible. Quenching offers the most inducements, for economic reasons, as when the copper is allowed to cool out the coppersmith has to cool out with it, it not being practicable to give him fresh work at each anneal; therefore, on these grounds, quenching for ordinary practice is preferable.

A few words now in regard to the process of annealing. Furnace annealing for copper pipes is excellent on paper and in theory. In practice it suggests many objections—namely, high first costs, a large, capacious furnace being requisite for the many lengths and varied-shaped pipes, heavy cost in upkeep in fuel, difficulty, perhaps not very great, of regulating temperature and securing uniform temperature throughout, also the distortions of the bends under heat. This latter is a great drawback, as even in steel pipes of sizes up to 5-inch bore $\frac{3}{16}$-inch thick, this takes place. There are equally good and more economical methods quite safe to adopt. Bends up to 5-inch bore can safely and satisfactorily be annealed on a large coke fire, with blast say, half on, and a covering of sheet iron over to retain heat. It is not at all a delicate operation, the heat being quite easily judged, and any tradesman of moderate ability can safely be trusted. The fool would scrap the work under any conditions. Pipes above 5-inch bore should be annealed by slinging them up with lower and horizontal within easy access of about a 2-inch blast pipe. Charge end of pipe nearest blast connection with light pieces of wood, lighting same with hot cinders or gas blow pipe; then blow blast through, the wood being repeatedly replenished. The effect to aim at is the emitting at opposite end of pipe of good supply of flame. Too much wood results in dense volumes of smoke, too little wood causes lack of flame and discharge of merely sparks and burnt wood. A little experience would soon enable the operator to decide upon the happy medium. A few minutes sees the pipe become a uniform red heat throughout its whole length. A 12-inch bore, or even larger bend, 12 feet or so long, can be heated by this means from end to end a good annealing heat in a quarter of

an hour or so, at an expenditure of a few coppers in wood, as old scrap template wood could be used. This method removes the possibility of any part not being properly heated. It may interest some to know that copper will attain to a degree of hardness by mere exposure to the atmosphere, the degree depending upon the time of exposure. It is advisable to re-anneal any pipes that have lain about for over twelve months, especially if any bending or working is required to be done to them.

Copper Pipes.—There are, generally speaking, two methods of making copper pipes—from seamless tube and from sheets brazed up. For all purposes where practicable, the seamless is the superior. The size of the pipe, radius of bends, thickness of metal and cost, are usually the determining factors. The cost affects the matter in this way: all seamless copper tubes above a certain size in diameter, and lighter in thickness than what is termed "basis price" size and gauge, rapidly increase in price per pound for each part of an inch larger in diameter or gauge lighter in thickness. It is, in the ultimate cost, more economical to make all pipes up to 6-inch bore, 12 W.G. thick, from seamless tubes, the ease and safety in bending the seamless tube more than compensating for the slightly larger outlay on a few of these tubes. The many-shaped pipes the coppersmith is called upon to bend make it bad practice to use seamed pipes where it is possible to avoid it, the seam often being a source of trouble, cracking and splitting when punished in the bending. When the size of the pipe or the smallness of the radius render it impossible to bend a tube satisfactorily, the pipes are built up from sheet, which method I shall deal with later. Any reference I make to tube or sheet only applies to them after delivery from the mills.

Seamless Pipes.—It may generally be taken as an axiom that copper which had withstood the test of being drawn into a tube has but little the matter with its quality. Any defects that subsequently arise are usually discovered during the bending and hammering, if not before, and they are purely

mechanical, but not connected with the metal itself. These defects, when they do arise, consist of longitudinal flaws or marks, detected upon the interior or exterior of the tube; generally the cause is carelessness with the dies in drawing the tube, dirt and grit getting between the steel dies and the copper. A further mechanical defect sometimes found is what is termed laminations, which are the result of blow holes in the original billets becoming rolled and drawn in the subsequent working, showing in the finished tube externally as scale or loose skin of copper, and internally as blisters. The presence of these defects, except in cases where the marks are very superficial, must mean the condemning of the tube. Tubes should be delivered soft, for economic reasons, and should be filled with resin and mixed with about 10 per cent. of pitch to prevent the resin crumbling. Lead should on no account be used. It is one of the worst of metals in its bad effect upon copper. It can never be absolutely cleared from the interior of pipes, even when the precaution is taken of whitening the inside of the pipe, and during subsequent heatings for annealing and other purposes the lead residue eats its way into the copper. To test the evil effects of lead upon copper, take a strip of the latter metal, and with a minute piece of lead upon it, heat over the fire to a good red heat, note the energetic action of the lead, and, after cooling, carefully examine the surface of the copper acted upon by the lead, remembering at the same time that the longer the copper is heated the deeper the lead will penetrate. This is the action which takes place upon the bends and other inaccessible parts of the pipes when heated after having been treated with lead. All bends in ordinary practice can be shaped on the resin. Apart from the disadvantages mentioned, lead is excessively heavy and, therefore, expensive to handle. The actual bending of the tubes suggests some interesting problems, and has considerably to do with the ultimate success or otherwise of the pipe. It is not necessary to demonstrate the obvious fact that bending a copper tube lengthens the distance between two given points at the back of the bend, and

reduces the distance in the throat of the bend. This naturally results in stretching the copper at the back and thickening or "puckering" the throat, the amount depending upon the radius. The thinning can, however, be seriously affected by wrong methods. The correct method of bending is to extend the area for thinning or stretching as much as possible, thus reducing it to a minimum in any given place. To do this, the blocks placed at the back of the pipe, across which the pipe forms a beam (the bending pressure being applied between) should be kept as widely apart as the radius of the bend required will admit, not less than four times the diameter of the pipe for the sharpest bend, and wider as the radius is increased. By this means it will be found that the tendency to reduce the thickness at the back of the bend is remarkably little, not exceeding one gauge in a fairly short radius. A radius of two and a half times the diameter of the pipe (to the center) is about as short a bend as is possible to execute satisfactorily. This applies to any size of pipe. It will be noticed that a copper pipe very rarely gives way across the bend when actually under working conditions; the tendency is for the pipe to split round the sides of the bends, due to the continual altering in shape from spherical to oval and back again when the bends are actuated upon by the expansion and contraction in the range of pipes. This tendency to split is enormously exaggerated in the case of a seamed pipe. Any weakness of the original tube in this direction will in all probability be discovered by the mechanical testing process previously referred to, of turning over a flange. The fact will be readily grasped that as the back of the bend stretches or lengthens the throat of the bend shortens, and the material has a tendency to jump or thicken up; the lighter the material the less the thickening takes place, and we find that the copper works up into folds or puckers on the inside radius of the bend; the lighter the material of the pipe and the shorter the radius the more it puckers. Now it is a great mistake to think that so long as these "puckers" can be beaten down with a hammer and leave all smooth, no harm has

been done. How often one finds after vigorously hammering down some specially large puckers that part of the bend where the puckers have been punished shows a series of irregular cracks, and has to be condemned. This is due to overstraining, an evil previously described. In this case, again, it is impossible to lay down hard and fast rules, but if the tradesman is aware of the possibilities of the evils he can, at least, err on the right side, until from experience he learns the exact amount the copper will stand. In further reference to hammering of copper bends, any tendency towards slovenliness in this respect, although perhaps not in itself anything more serious than being unsightly, should cause grave suspicion of like carelessness in more important matters.

Flanging.—The flanging of copper pipes is an important process, but with reasonable care and supervision can be rendered simple and trustworthy. The flanges should be cast with a collar two-thirds in height of flange thickness. They are usually faced and drilled before fixing on the pipe, and should be cast with a decided taper to within $\frac{1}{8}$ inch of flange face, say a taper of $\frac{1}{8}$ inch, to ensure easy access of the solder between flange and pipe. It is of the utmost importance that inside of flange and outside of pipe ends be thoroughly filed clean, as dirt is the great enemy to a good braze. To fix flange it is only necessary to calk over the copper on the flange face to keep it in position. Any knocking out of the copper pipe from inside at back of collar (sometimes done by incapable mechanics to keep flange in position) may be set down as another evidence of slovenly work, and condemned. Under no conceivable circumstances is it necessary, yet one occasionally comes across some deplorable cases. The actual brazing on the flange is often rendered insecure by bad methods. It is a common practice to put the flange over the fire and blow up to a red heat, then cut off the blast and charge with solder, then blow up again until fluxing takes place; and thus previous to the charging with solder the collar top and the space between pipe and flange has become covered with dust from the fire and the flange and pipe covered with scale, rendering

it difficult to make a good, sound braze; sometimes powdered borax is sprinkled on, but it is unsatisfactory. The correct method is to carefully wash the brazing solder, which should be coarse (No. 6), in clear water, then mix a handful of granulated borax with the wet solder and charge flange before putting on the fire; the moist borax runs into space between flange and pipe and keeps it chemically clean, and no trouble is experienced in making a good, sound joint. Extra charging when flange is at brazing heat should be avoided, but if necessitated by flange collar not being sufficiently filled the extra solder can be applied without the slightest need for stopping the blast, for the longer the pipe remains under heat the greater the effect of damaging gases and the more the molten solder penetrates into and reduces the thickness of the pipe. This latter effect is the cause of many pipes prematurely "necking" at the flange. Many coppersmiths complain that under the method I put forward the solder becomes "drossy." This condition is not due to the method adopted, but to the solder not being sufficiently coarse. The brazing, or the assisting of the brazing, of flanges by gas blowpipe is sometimes recommended. The use of the blowpipe for brazing may be classed as a necessary evil, that is to say, its use is pardonable only in extreme cases, for work inaccessible on the fire, or as an emergency, to save a flange giving trouble by the brazing solder running through. I condemn flange brazing by means of blowpipes on several grounds:

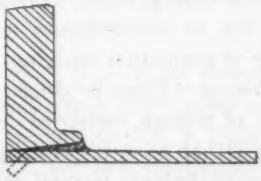


Fig. 2.—Test for Presence of Solder.

1st. The deleterious effect upon the solder of a fierce flame, causing what coppersmiths term "perishing." This is shown by the extraordinary amount of borax necessary to flux the

solder, and the "drossy" condition of the solder after a few minutes' action upon it of the blowpipe flame.

2d. The fierce play of the blowpipe flame upon that part of the copper immediately above the flange, resulting in excessive and more or less prolonged heating. This part of the pipe should be particularly guarded from any prolonged heat.

3d. The cost of blowpipe work is excessive, and if gas expenditure is charged against the shop expenses it will tell its own tale in a few months.

It would be imperative to make an outlay in blowpipes for each fire used, as if not, delay would ensue from men waiting about for turns in the use of blowpipes. There are no advantages in blowpipe brazing excepting in the emergencies stated, or for use in emptying the resin out of large, unwieldy pipes, and I should call a blowpipe the incapable man's friend. The blowpipe is no new thing and has been used for intricate work for twenty or thirty years. The pipe must be hoisted from the fire as smartly as possible after fluxing takes place. Smartness is the keynote to all successful brazing, as no time whatever must be lost during the process. This applies to all descriptions of brazing. The indications of a good flange braze are neatness and clearness round solder on the collar, and evidences of solder when any part of the flange calking is set back a little from the flange face with a blunt chisel within at most an eighth of an inch; this test would in no way damage the pipe as per sketch.

There should be no filing round brazing at the collar, unless with a round file, as the tendency to file square corners is a frequent cause of premature necking.

Copper Pipes Brazed.—Pipes produced from sheet offer the greatest possibility of failure, owing to defective work. The pipes can be made either as straight lengths and afterwards bent, or shaped up in halves, termed saddle and back, and brazed together. The size of pipe, radius of bends and thickness of material again determine the process. The making up into lengths and bending on the resin is the less costly, and can be adopted from any sizes, say from 6-inch bore up

to 12-inch or even 14-inch bore, providing, of course, a press of sufficient power is available for the purpose; the larger sizes would require a hydraulic press of 60 to 80 tons capacity. The straight length is produced by taking a sheet equal in width to the circumference of the diameter of pipe bore required plus $\frac{1}{4}$ inch. Thus for a 10-inch bore pipe 10 W.G. thick, take the circumference of $10\frac{1}{4} = 32$ inches; for $\frac{1}{4}$-inch thick pipe take circumference of $10\frac{3}{8}$ inches. This sheet is thinned along each edge, wedge shape, by hammer or rolls. This sheet is afterwards bent up, pear shape, to offer a greater resistance to bending (see Fig. 3) and springing of the seam when only one side is being heated, as during brazing, also by having the seam at the small end of the pear-shape section a small surface is presented for the solder to spread over, making a neater and better seam, as per sketch.

The sheet is annealed and well cleaned along the thinned edges. The copper sheets should be thinned to the width of four times the thickness of the material, and one edge cut down with a sharp chisel to a depth of $\frac{1}{8}$ inch more than the

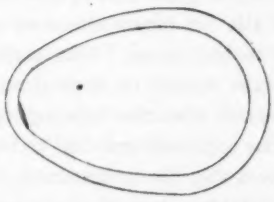


Fig. 3.—Section of Straight Pipe from Sheet Prepared for Brazing.

thinning, about 1 to $\frac{1}{2}$ inch from either end, to form the cramp. These figures apply to all pipes and for any purpose or pressure. The cramp is cut $\frac{1}{8}$ inch deeper than the thinned part to allow for any springing during brazing, as shown in the small specimens before you. The idea of the cramp being out will probably be fairly clear, as without the cramp it would be impossible to regulate the amount of lap. The edge of the side not cut into is placed between the cramp

and the whole drawn together and bound with soft iron wire about $\frac{3}{16}$-inch thick. The pipe is placed on a mandrel not exceeding 3 inches diameter, and hammered down close along the seam. The pipe is now slung up ready for brazing, and, to insure perfect cleanliness and facilitate the solder fluxing easily through the seam, a solution of boiling-hot water in which borax has been dissolved should be poured along the seam. About a handful of borax in a pint of water will usually suffice. This will remove any greasiness or foreign matter from along the part to be brazed. The seam is loaded with coarse solder, and only experience will determine the quantity to use. The best plan is to have a length of small-section angle iron or copper shaped similarly some 10 feet or so in length, and charge this with solder uniformly the whole length, place the charged angle iron along the seam, inside the pipe, and turn it over and empty the solder on to the seam. It is advisable to blow up the fire before putting the pipes on. This may seem an unimportant matter, but it is the small things that many neglect and which often make all the difference between a good and a moderate piece of work. The blowing up of the fire clears the dust and dirt out, without blowing it on to the seam. The seam is drawn slowly across the fire to braze it, and no time should be lost between the seam being cleaned and the brazing of it, otherwise the edges rapidly become oxidized and bad to braze cleanly. The rate of progress across the fire is determined by the fluxing of the solder. The outside edge of the seam, which tends to spring open under the heat, is put up close by means of the pressure from a flat-ended iron bar. If the seam has been properly cleaned and prepared no difficulty is experienced in steadily and regularly working across the fire. Sometimes, however, the careless operator has not sufficiently cleaned his seam, and he has a difficulty in getting parts of it to "take," as it is termed, his solder fails to flux properly and runs "greasy." If not prevented, the coppersmith who may be incapable will keep his pipe stationary on the fire while he is trying to persuade the solder to unite to the copper. Now,

bearing in mind the comments made in the section upon annealing, of the disastrous effects of prolonged heating, it can readily be gathered how fatal it will be to the seam to allow it to remain for long over the fire. In cases of this kind the parts giving trouble should be passed over and progress continued with the remainder of the seam. After removing from fire and cooling off, the defective places should be thoroughly scraped and cleaned both inside and out, and steeped in a solution of vitriol to remove the burnt borax and other dirt, then returned to the fire and rebrazed. This may be a little trouble, but it is imperative. The necessity for cleanliness cannot be exaggerated, and slackness in this respect is responsible for the majority of failures in brazing. It will have been noted, no doubt, that solder in brazing unites to the copper, that is, it penetrates and becomes part of it; but the copper must be thoroughly clean for it to do so, if not, the solder merely flows over the copper, and to the inexperienced observer appears perfectly sound, whereas the slightest strain will part the two metals. Many in dressing off the exterior of the seam furrow the copper with a file, or make sharp corners; both are harmful and unsightly. A good seam can be determined by the closeness of the outside edge of copper, the absence of any groove, hollow, or square edge along the seam, the presence of a regular straw-colored seam of solder along edge, and no evidence of thickened edge. Pipes made up in this way should not be hammered if for bending, but only malleted; hammering, if in any way extensive on the seam, reduces its bending stress.

Copper Pipes Worked Up.—The built-up pipes, saddle and back, are by far the most interesting, both from the point of view of the mechanic and the observer. The same rule as to width of sheets applies in this case as in the straight length, only the sheet is cut in halves in width for saddle half and back half. In shaping the copper correctly there are a few well-defined rules to bear in mind, applicable to all shapes, sizes and radii, which, if carefully followed, will result in a first-class production, with the copper unimpaired, if not im-

proved in ductility. It is hardly necessary to point out that without the requisite skill, which can come only by practice and general experience, all the reading and studying possible will not enable a good piece of work to be produced. I cannot, by reason of time limit, give every detail of the processes; a word-picture of such would be totally inadequate. The fact must be grasped that copper possesses the property of flowing, if correctly handled; that is, the thickness of the metal can to a large extent be retained, and the particles of copper worked to different positions. To give one example typical of this principle, lay down a full-size sketch of, say, a 10-inch bore right-angle bend, say 15-inch radius to center of bend. If the sketch be carefully measured round, the center radius will be found to be 50 per cent. greater in length than the inside radius. Thus theoretically a piece of copper worked into shape should be much thinner round the center radius part than, say, along the throat. Say if .160 inch or 8 W.G. to begin with, should finish round center radius .104 or 12 W.G. If .128 inch or 10 W.G. to begin, should finish round center radius .080 inch or 14 W.G. After, however, the copper has been made to flow by right methods, it can be worked over with a loss not exceeding one gauge in thickness, that is for any reasonable radius. The copper must be worked over in regular courses, not exceeding 1½-inch pitch.

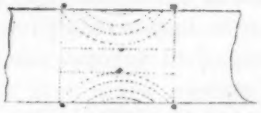


Fig. 4.—SHOWING METHOD OF MARKING OFF SHEET FOR WORKING OVER SADDLE OF BEND.

The object of the regular courses, as shown by dotted lines on sketch, is to gradually force the particles of copper to the edge, and also, in all working or "razing" of copper, the regular coursing keeps the shape of the work to the radius desired. The sheet, being marked off as shown on sketch No. 4, is held at an angle of not more than 10 degrees to block or

anvil, and commencing at point *A* work to *B* and back to *C*, then from point *A* to *C* and back to *B*, working alternately in courses of not exceeding 1¼ pitch. The copper to be only beaten down to the anvil, not on any account to be given a solid blow upon the anvil. When worked to edge, opposite side of sheet to be treated similarly, and then the whole annealed and hammering repeated for some five times for an ordinary radius—the finishing of shape can be done on an iron mandrel. The back, as will readily be seen, is contrary to the saddle, and shorter round center radius than outside radius. The material will gather round center radius and thicken; this must be beaten out.

Fig. 5.—Showing Method of Marking Off Back of Bend.

The back sheet is hollowed out by marking, as per sketch No. 5, and regularly hammered with wooden mallet on dotted lines, blows not repeated in one place or undue straining results, then annealed, and hammering repeated until copper is up to the template. The sides of the back of a fairly sharp bend should be "razed" in on an iron "cod" with a full-faced hammer to assist the shaping. If the halves after shaping are placed upon a full-sized sketch it will be found that the saddle is short of the center mark and the back in excess, that is, if correctly worked. These halves are thinned along edges as in the straight lengths, and cramps cut in after annealing and carefully cleaning. The cramps must not be cut in the radius or bend of the pipe, but in the straight; if cut in bend the pipe is likely to split at the cramp during the altering in shape of the bend under actual working conditions. Cramps are better cut in the backs to facilitate dressing off of the seam after brazing. I have made no mention of any test for copper sheet; I must confess to having no faith in any mechanical testing. The great majority of bends in

ordinary practice, if correct methods are adopted, put little strain upon the copper, and ordinary commercial copper is quite capable of meeting the requirements. Exceptional copper work requires special pure copper to stand the manipulating, and carries with it, also, a special price. It is rarely that firms of good repute supply bad copper sheet from their mills; when they do, all the bending tests applied will not, I fear, determine its quality. There does not appear to be that coöperation between the chemist or metallurgist and the actual worker of the copper without which it is impossible to arrive at any trustworthy data. The connection in this direction is not nearly so complete as in the case of iron and steel. Many times the coppersmith has supplied to him copper sheet of abnormal hardness, that is, sheeting which, despite the most careful annealing, never reaches the normal state of ductility. The man working the sheet is soon conscious of this fact, and exceptional care is required not to overstrain it. The difficulty in respect to this phenomena is, that if the copper be tested for tensile stress it will give a higher figure than the more ductile sample. I can offer no explanation of this; mere theory, without some reasonable substantiation, is useless. Perhaps the analyst could assist in this important matter. Some of the copper people can supply a really good soft copper, but at a rather hard price. I believe the best to be electrically pure copper deposited and then put through the melting process and afterwards rolled in the ordinary way. Superficial marks or roughness in copper sheets are, unless deep, of little moment, the character of the copper itself is the most important. It may be of interest to notice that any impurities contained in copper come to surface hammered upon, if worked sufficiently. It causes minute cracks to appear, which, if not removed, will soon work through the material. To prove that it is due to impurities it is only necessary to point out that these cracks are usually at first isolated, and when only on the immediate surface can be cut out by a gouge chisel and they will not again show themselves in subsequent working; also the small

pieces cut out are, upon examination, found to be foreign substance, excessively hard and discolored similar to what is sometimes termed "dross."

Hydraulic Testing.—It is essential in pipes intended for pressure, and in seamed pipes, to test them with hydraulic pressure to detect leakages. In doing this care must be taken that the jointing for testing is kept outside the copper pipe on the flange face, otherwise it will not be a genuine test of the flange brazing.

To exclude the air from the pipes they must gradually rise the whole of the length; it is not sufficient to have the exit the highest point; if any other part varies in height the air will remain at that high point and cannot possibly be forced past the depression (see sketch No. 6).

Fig. 6.

To test the presence of air in any part of the pipe two methods can be adopted and neither will give rise to any suspicion of doubt. If when the pressure is turned on after the pipes are filled the pointers of the pressure gauge immediately and steadily advance, no air is in the pipes; if they refuse to move at once or violently oscillate, that indicates the presence of air. If the highest or any other point in the range of pipes be given a sharp rap with a hammer, the pointer of the gauge will reciprocate and vibrate, that is, if no air present. Air being present, they will fail to respond. Unfortunately, hydraulic testing is of little use in determining the strength of a pipe; only by a careful use of data, based upon long experience, can the necessary scantling be fixed, and if the method herein described is adhered to, no misgivings in regard to workmanship need arise.

Design of Pipes.—In the designing of the copper pipes by the draughtsman much can be done to assist the expansion

and contraction effects. Space and general arrangement enters largely into the question of pipe design, and is undoubtedly restrictive. A few points may be mentioned to be borne in mind in fixing up the ranges of piping. As much straight as possible to be left at the flange ends of pipe; if practicable not less than 4 or 5 inches, double that where permissible. All radii to be kept as large as space will allow, to avoid undue stiffness and to reduce friction of the steam and water. Most designers and draughtsmen today are quite capable, if permitted, of laying down a good workable pipe arrangement, and generally show themselves quite conversant with up-to-date practice.

Generally.—I am not altogether unconscious of the fact that many coppersmiths lay the blame for failures upon the draughtsman, or may be the copper manufacturer. This, however, is not peculiar only to coppersmiths, and seems to follow the line of least resistance in many cases. I scarcely know what to answer to the critics who condemn present-day copper and workmanship and eulogize that produced in their youth. Probably, as in other matters, "distance lends enchantment to the view;" at any rate no attempt is ever made to bring facts to prove the assumption. Most of us are familiar with the tendency of age to, by constant repetition, keep fresh the victories, and by careful respect deface from memory the failures. Personally, I find much the same conditions in regard to the quality of copper today as, at any rate, twenty years ago, but better results are produced by more up-to-date and systematic methods available now. The substitution of steel for copper in cases of high steam pressure and other directions has developed rapidly since the production of the seamless steel tube—a distinct improvement certainly when correctly designed, and adequate provisions made for its characteristic action. A treatise upon the working of steel piping may be useful at some future date, but with all this substitution of steel the doom of the copper pipe is not pronounced yet. Naturally I have a great faith in copper work, and resent strongly any insinuation that the copper-

smith has deteriorated. We possess on the northeast coast, amongst our coppersmiths, tradesmen second to none in the world for either ability or despatch, and who can be depended upon under favorable conditions to produce first-class work. If engineers will closely examine the matter I have placed before them and memorize the main points, they can feel assured of being quite able to determine from stage to stage in the productions the actual quality of the pipes supplied.

THE ENGINEERING OF ORDNANCE.

By A. Trevor Dawson, M. Inst. C. E., M. I. Mech. E.*

PROBLEMS IN THE MANUFACTURE OF STEEL FOR GUNS.

The manufacture of the large gun tubes of today, especially when using the high-tensile alloy steel introduced during the past few years, involves the solution of many difficult problems, of which the three principal may be said to be—

1. The ensurance that large hollow forgings shall throughout be regular and even in mechanical properties.

2. The overcoming of mechanical and physical difficulties attached to the production of large masses of steel free of cracks, flaws, sandy defects, "ghosts," etc.

3. The production of considerable weights of steel of high tensile strength and elastic limit, but showing no trace of brittleness.

The first problem has now been disposed of as a result of experimental work on a large scale. It might at first sight appear that the production of large masses of steel in the form of an ingot, and of absolutely even composition throughout, should present no great difficulty, but it has been found absolutely impossible, owing to the irresistible tendency of certain elements in the steel to segregate; three elements, carbon, sulphur and phosphorus, show this migratory propensity. They are always found in the greatest quantity in the part of the ingot last to solidify.

It is absolutely necessary that no such segregated area shall become part of a gun tube owing to the inferior properties of steel containing such impurities, and in the absence of any

* From the Gustave Canet Lecture, delivered at the 25th Anniversary meeting of the Junior Institution of Engineers.

method of preventing such a segregation a great deal of research has been carried on for years in order to find exactly in what manner segregation takes place. By suitably arranging the ingot mold it becomes possible to cut away the impure portion without spoiling the remainder for gun-tube manufacture.

Latterly a new means of detecting sulphur segregation has been put into extensive use. This method consists of sectioning the steel forging to be examined, polishing it and squeegeeing down on the polished surface a sheet of ordinary photographic bromide paper soaked in weak (about 10 per cent.) sulphuric acid. Interaction takes place between the manganese sulphide of the steel and the sulphuric acid, and gaseous sulphuretted hydrogen is produced. This in turn interacts with the silver bromide in the paper and forms a brown insoluble deposit of silver sulphide. In this way each particle of manganese sulphide existing in the steel is clearly shown on the paper by a brown speck, and from the number of the brown specks the most accurate knowledge is obtained of the extent and degree of the sulphur segregation.

Fig. 8 is made from a sulphur print taken in this way from the longitudinal section of a 12-inch octagonal ingot, from which it is seen that the point of maximum segregation is on the central axis of the ingot and quite near the top. Were this ingot to be trepanned along its central axis all the segregation would be removed, and the remainder of the steel would be practically uniform in sulphur contents. As it has also been found that the other two elements mentioned, carbon and phosphorus, follow the same direction in segregating as does sulphur, it is evident that no appreciable inequalities in composition would be found after the removal of the central core. The methods employed in casting this small ingot and ensuring the location of the center of segregation in the head are applied at the Vickers works in the largest ingots, with exactly similar results.

The second problem mentioned, that is, the overcoming of mechanical and physical defects, is far too large a subject to

be dealt with in a short time, but, as in the case of segregation, it has been largely solved by the application of scientific and accurate methods of investigation.

Fig. 8.—Photograph of a Contact Sulphur Print From Section of a 12-Inch Octagonal Ingot.

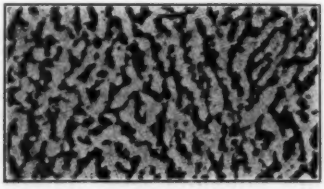


Fig. 9.—Nickel Chrome Gun Steel in "Pearlitic" State—Elastic Limit = 20 Tons per Square Inch.

Fig. 10.—Nickel Chrome Gun Steel in "Sorbitic" Condition—Elastic Limit = 35 to 40 Tons per Square Inch.

The third problem, associated with the production of a steel of suitable mechanical properties, has been met by the addition to the steel of suitable elements, notably nickel and chromium, and by the scientific heat treatment of the finished forging. In recent years a great amount of work has been done in this direction, and it is now found possible, by a correct sequence of heat treatments, very considerably to increase the tensile strength of the forging without in any way diminishing, and in fact increasing, its ductility and power to resist shock.

To show the very different mechanical tests in an alloy steel which can be produced by varying treatments, I show first on Fig. 9 a micro-photograph of a structure known as "pearlitic" produced by annealing, the steel in this condition being soft and giving an elastic limit of only 20 tons per square inch and a proportionately low breaking stress.

The other micro-photograph, Fig. 10, is from the same forging at a later stage, and illustrates what is now termed a "sorbitic" structure. This is the quality universally found in a finished gun tube. In this condition the steel shows, as compared with the "pearlitic" condition, a greatly increased tensile strength and as high as 40 tons per square inch elastic limit, with only slightly reduced resistance to shock.

I think these few slight indications of the trend of present-day metallurgy will serve to show the care each of the several parts of a gun receives during manufacture, and the anxiety that is shown in avoiding anything but the best possible and most suitable material for this work.

WIRE-WOUND VERSUS SOLID-STEEL SYSTEMS OF GUN CONSTRUCTION.

This consideration of the properties of modern steels naturally leads to a discussion of the two distinctive methods of gun construction—the one in which wire winding is a prominent feature as adopted in the British service, and the solid-steel system identified with guns for continental powers.

The chief advantage of the wire-winding system is that uniformity of stress is attainable throughout the whole of the material employed in the gun's structure to an extent that is impossible in the case of a gun built up of steel hoops only. Weight for weight, the maximum stress being the same, the gun constructed of steel and wire is therefore of the more efficient construction. This uniformity of stress throughout the whole mass of the gun is due to the wire being wound on at the theoretical tension necessary to obtain from every layer the maximum resistance when the gun is fired. Moreover,

the breaking stress of the wire used is about twice as great as that of the best steel which it is possible to have for solid-steel construction. Again, by the regulation of the initial tension on the successive layers of wire during construction the full effect of the reinforcement is obtained to a degree that it is not possible to get by shrinkage, which is a difficult operation, especially in the case of long hoops. The wire is much more likely to be free even from minor defects, firstly, because of the process of manufacture, and, secondly, because it is possible to inspect and test the wire throughout its complete length.

If the inner A tube of a wire-constructed gun fails, it is still possible to continue firing without danger, while with a gun constructed entirely of steel the weapon is rendered useless in the event of the inner tube splitting.

If in a tube or hoop there be a flaw, whether hidden or visible, there is great danger of such flaw extending until rupture takes place. On the other hand, where wire is used, a rupture in any coil cannot spread to adjacent coils. Should a large-capacity high-explosive shell burst in the bore the wire construction will prevent the explosion causing serious damage to the turret. The construction utilizing steel and wire enables the gun to be relined and rendered nearly as good as new. This is more difficult in guns of solid-steel construction. It will therefore be seen that the gun of wire construction is stronger and better able to resist the great radial stresses due to firing than a gun of solid-steel construction of the same weight. It is safer in the event of accident and more easily repaired.

In some of the continental countries, however, the solid-steel construction is preferred; the various points on this subject were discussed in Mr. Canet's Presidential address to the Junior Institution of Engineers, and the principal reason given by him in favor of the solid-steel construction is the recent great advance made in the provision of steel in large homogeneous pieces of splendid quality, high elastic limit,

great breaking strain, and with great elongation. In view of these advantages Mr. Canet considered that wire construction was not necessary.

THE CONSTRUCTION OF GUNS.

The first process in manufacture, as carried out at the River Don Works, is the casting of the ingot in a mold, generally octagonal. Through the ingot a hole is trepanned from end to end, an operation which is carried on in a boring mill. The trepanning bar varies in diameter from 15 inches to 21 inches, according to the ultimate internal measurement of the gun or jacket. The machines in use are capable of taking an ingot of over 15 feet long, the total length of the machine being over 40 feet.

The ingot is next forged under one of the hydraulic presses. A hollow mandrel is used, so that continuous supply of water may be forced through to prevent it from becoming over heated while the great red-hot mass of steel is being gradually worked to approximate length and diameter. The presses are worked by a special arrangement, which enables the engine to run continuously, although the press is intermittent in its operations. With this combination a very quick action is secured; the engine, continuously running, is always ready to put on the maximum pressure when the actual operation of forging begins; whereas, with the usual simple direct operation time was lost, as the engine had to reach full power from low speed or from rest. The intermediate pressure of 1,120 pounds is also utilized for the quick return of the main ram. Furnaces adjacent to the press are used for reheating the forging.

The gun tube forged to approximate dimensions is next passed to the lathes and other machine tools, some of them being 90 feet in length, with headstocks of proportionate diameter, and all driven by independent electric motors. The forging, having been rough-turned externally, is bored, and some of the machines utilized for this purpose have beds

108 feet in length, the work in some cases being carried on simultaneously from both ends.

The tubes are next reheated and tempered or hardened in a bath of oil. The heating and oil baths are under the ground level, and the large Goliath crane lowers the gun tubes into the baths. Test pieces are then taken from both ends in order to ascertain if the forging is satisfactory before being finally machined.

Wire winding is one of the most interesting processes in connection with the manufacture of modern guns. Round the tube, which has been rough-bored, there is wound steel wire or ribbon, about $\frac{1}{4}$ inch wide and $\frac{1}{16}$ inch thick, and of a tensile strength of 100 tons to the square inch. The wire is put on at high tension. The reel, with special gear for ensuring that the wire will always be at the correct tension, is mounted on an extension of the saddle of a large lathe, on which the gun tube is rotated. The end of the wire or tape, having first been secured to the tube at the breech end by being wedged into a small recess, the gun is rotated, and the wire tape wound upon it at high tension. The saddle with the reel is traversed by screw-gear along the bed of the lathe, so as to wind the wire in one close, continuous layer from breech to muzzle, and *vice versa*, until the required amount has been wound upon the tube. The tension of the wire is varied with each successive layer. This tension is insured in a simple way; the wire or ribbon, as it is unwound from the reel, passes to the gun tube through dies of hardened steel, with slightly curved surfaces. These dies or nipping blocks are pressed together by a series of levers and weight, in order to obtain the necessary pressure. Any variation is possible by changing the weight on the levers.

After the gun is wire wound it is, if necessary, skimmed up in a lathe to get rid of any irregularities due to the difference in the section of the wire. It is then carefully measured, and the internal diameters of the outer tubes or jackets are carefully bored to suit. The building up of the gun is carried out at the Goliath crane, to which reference has already been

made. It lifts 100 tons, travels on rails laid for a length of 200 feet on a gauge of 55 feet, over the oil baths and heating furnaces under the ground level. The height from ground level to the hook at its highest position is 77 feet, so that the longest guns made can be carried.

The final machining of the guns externally is the next operation.

The breech bush which carries the block and the breech mechanism has meanwhile been manufactured. The machines used are mostly of special design to ensure extreme accuracy in every dimension, and all are electrically driven. At almost every stage tests are made and the work very carefully inspected, and when the gun is completed a final inspection is made before the weapon is sent to the range for firing trials.

POWDERS AND EXPLOSIVE COMPOUNDS AND THEIR INFLUENCE ON THE BALLISTICS OF GUNS.

The second influence on the development of ordnance—progress in powder and explosive compounds—has been of even greater importance than metallurgy, especially in the designs and velocities attained in guns.

The comparatively recent introduction of smokeless powder into our service and its continually-increasing efficiency has, perhaps, influenced the type and construction of ordnance more than any other factor. France was the first country to adopt a smokeless powder for ammunition for use with the Lebel magazine rifle; that was in 1885. Since that date many preparations, giving little or no smoke, have been proposed as substitutes for black powder. These consist essentially of guncotton or nitrated celluloses. To Mr. Alfred Nobel, the inventor of dynamite, is due the combination of nitroglycerine with guncotton for the production of a smokeless explosive. Camphor was employed by Mr. Nobel as an agent for promoting the union of the two explosives, and the result was called ballistite. Camphor, however, does not

remain a constant ingredient, and its evaporation, even at ordinary temperatures, leads, in course of time, to alterations in the ballistic properties of the material. This would be a grave defect in any explosive, especially when it has to stand exposure in various climates, and the defect was eventually recognized. It was with the view of removing the defect in question that the extensive experiments of Sir Frederick Abel were carried out, the result of his labors being the explosive known as cordite. The experience of some years has proved that these powders can withstand severe climatic changes without serious deterioration, chemical or ballistic.

Much higher ballistics became possible with the new powder, owing to the fact that a much larger total volume of gas is developed for the weight of charge, and therefore greater velocity of the projectile is attained than formerly, the gas production taking place gradually during the whole time of the passage of the projectile down the bore, with a correspondingly more uniform distribution of pressure. The total propelling force is greater, but as it is more regularly sustained the maximum pressure is not greatly increased, if at all. It follows that for the same or even lower maximum pressures higher velocities are obtained.

The composition of cordite is as follows:

Nitroglycerine, . . .	58 per cent.
Guncotton, ...	37 per cent.
Mineral jelly, . . .	5 per cent.

The composition of cordite M.D., the latest service explosive, is:

Guncotton, ...	58 per cent.
Mineral jelly, . . .	5 per cent.
Nitroglycerine, . . .	37 per cent.

The explosive compounds for artillery purposes now used by the principal Governments of the world may be broadly divided into two classes, namely: "Nitrocellulose" powders and "Nitroglycerine" powders. The word powder is mis-

leading as applied to modern propellants, in that they are made in a variety of forms which do not bear any resemblance to a powder, but take the shape of tubes, cord, strips, flakes or cubes.

"Nitrocellulose propellants" consist of nitrated cotton, mainly of the soluble form, gelatinized by incorporation in a kneader by a suitable solvent, generally consisting of a mixture of ether and alcohol, and then pressed or rolled and cut into a suitable form, dependent upon the nature of the gun for which it is intended. Some Governments which use nitrocellulose propellants incorporate therein certain chemicals with a view to improving the keeping qualities or to indicate, by a variation in the color, if any change occurs in the condition of the nitrocellulose.

"Nitroglycerine propellants" consist of a mixture of nitrocotton, mainly of the insoluble type (*i. e.*, guncotton), and nitroglycerine in varying proportions, with or without the addition of certain bodies, such as mineral jelly, which has a waterproofing effect, thus protecting the powder against the deleterious influence of moisture in the atmosphere. The mass is incorporated in a kneader, together with a solvent, usually acetone, and then pressed or rolled and cut into suitable forms in the same manner as nitrocellulose propellants. It has latterly been found that if, in addition to the mineral jelly, which exerts principally a physical action, a very small percentage of certain alkalis are also incorporated in the propellant, the stability is very greatly enhanced.

As regards high explosives for loading into shells, picric acid, until comparatively recent times, formed the basis of the explosive adopted by the majority of countries throughout the world.

Lyddite, melinite, shimose powder and ecrasite consist, for the most part, of picric acid; in fact, lyddite is simply melted picric acid. Latterly, however, with the exception of Great Britain, the majority of nations have discarded the use of picric acid and adopted instead trinitrotoluol, which, though only very slightly less powerful than picric acid, possesses

many advantages over it, one of which is that it will not form dangerous combinations with other bodies with which it may come in contact, whereas picric acid in contact with certain metals forms very unstable and dangerous picrates. Moreover, trinitrotoluol is not unpleasant to work with, nor does it disfigure, like picric acid.

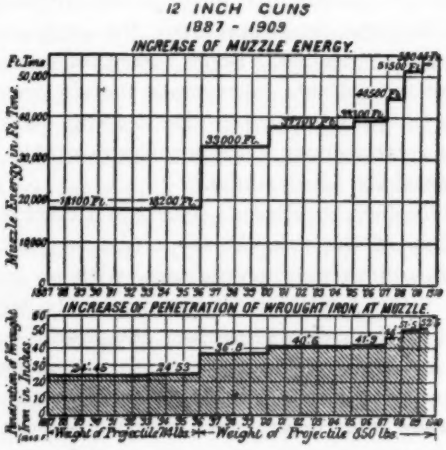
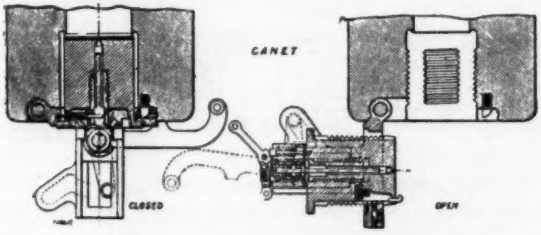


Fig. 16.

Fig. 16 clearly indicates the development in energies due to the application of modern explosives of more uniform distribution of pressures. Twenty years ago, before the introduction of nitrated powders, the barrel of the 12-inch gun was only 27 feet long and the velocity attained 1,892 feet per second, giving a muzzle energy of 18,100 foot tons. The length has been gradually increased to 51.5 feet, the muzzle velocity to 3,000 feet per second and the energy to 53,045 foot tons, as shown on the diagram. The penetration through wrought iron has, therefore, increased from 24.5 inches to 52.5 inches, as shown on the diagram.

CANET BREECH MECHANISM.

Figs. 18 and 19 show the breech mechanism designed by M. Canet. The breech block is made with four sectors of interrupted screw threads, and is supported by a tapered ring, which fits into a recess in the breech of the gun. The ring is either locked to the breech of the gun when the breech is closed, or to the block when the latter is withdrawn from its seat, by a sliding catch recessed into the breech. The ring is made in one piece along with the sliding bracket, and hinges on the left-hand side of the gun.



Figs. 18 and 19.—CANET BREECH MECHANISM.

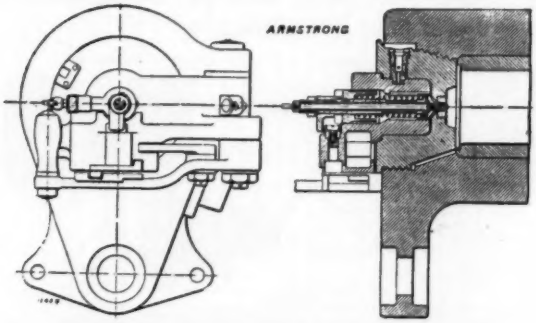
For opening and closing the breech the back of the block has bevel teeth, which engage with the bevel wheel on the hand lever. At the end of the spindle is a lever, provided with a roller, which works in a horizontal cam in the bracket. On pulling the hand lever the block is first rotated one-eighth of a turn and, by a continued pull on the lever the block, sliding back on the bracket, is withdrawn from the breech. The block, with bracket, then turns upon its hinge free of the breech of the gun. The cartridge case is extracted by two hooks, held up to their work by spiral springs.

The gun can be fired either by percussion or electricity. To ensure that the gun cannot be fired before the breech block is fully screwed home, the pin of the percussion gear is held by a spring "catch," which prevents it being cocked until the catch enters a recess in the breech block, and this is only possible when the block is in its locked position.

In the electric-firing arrangement no contact is obtained before the gun is closed, so that the gun cannot be fired prematurely.

"ELSWICK" SINGLE-MOTION BREECH MECHANISM.

An Elswick single-motion breech mechanism for a quick-firing gun using metal cartridge cases is illustrated by Figs. 20 and 21. The breech screw has interrupted threads divided

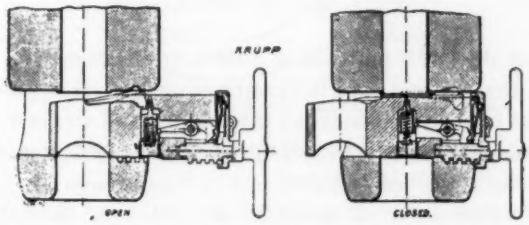


Figs. 20 and 21.—ARMSTRONG BREECH MECHANISM.

longitudinally into two parts, the front portion being conical and the rear portion parallel. The threads in the front portion are opposite interruptions in the rear portion for the purpose of equalizing the firing stresses on the metal of the breech surrounding the screw. The breech screw is mounted on a carrier hinged at the right-hand side of the gun, and is secured directly to the carrier by a retaining screw. The arrangement for opening and closing the breech mechanism by a single motion consists of a block mounted in the carrier and capable of sliding laterally across the rear face of the breech screw, the latter engaging with it by means of a projecting pin. The block is connected to the hand lever by an intermediate link. This opening gear is arranged so as to give the greatest power when locking or unlocking the breech screw.

The firing gear is of the combined electrical and percussion type, and is mounted in an axial hole bored in the carrier and breech screw and secured by a nut in the rear end of the carrier. The striker is actuated by a powerful spring which surrounds it. A safety device is mounted on the rear face of the carrier and consists of a vertical spindle having one toe in engagement with a downward projection on the striker and, on its lower extremity, two arms, one in engagement with the groove in the hand lever by which it is actuated, and the other with a groove in the sliding block by which it is retained when disengaged from the hand lever. A retaining catch in the carrier holds the breech screw in the open position until the mechanism is returned to the face of the breech. The cartridge extractor consists of a horizontal spindle, shaped on its inner end to take the rim of the cartridge case, and having a lever mounted on it at the exterior of the gun. One end of this lever has a spring plunger to retain it in its normal position, and the other portion has pivoted to it a bar mounted in the lugs on the breech ring and engaging with an eccentric path on the carrier hinge.

KRUPP BREECH MECHANISM.



Figs. 22 and 23.—Krupp Breech Mechanism.

Figs. 22 and 23 show the Krupp mechanism, which is quite different from those just described, as the block is of the wedge type. For this wedge the breech of the gun is slotted out at right angles to the bore, the opening so formed, having a flat face and a semicircular back, is made with a

taper enlarging from left to right. The breech block fits the slot and is provided with horizontal guides parallel with the back of the slot. At the large end of the block is the locking screw with a handle which engages with threads cut in the breech whereby the block is locked to the breech when in position. The firing pin is cocked by means of the lever actuated upon the locking screw while the breech block is unlocked. The cocking lever cannot be released or the gun fired before the breech is closed, as the safety gear is held up to its work by a flange on the locking screw. This flange is interrupted in one place where the gear can be pulled back to release the cocking lever; but this is possible only when the locking screw is fully screwed home and the breech locked.

The mechanism is fitted with an extractor for withdrawing the empty case. The extractor is actuated by the breech block at the end of its travel. The opening of the mechanism is achieved by turning the handle to the left, and simultaneously the cocking lever carries the firing pin rearwards, and the threads of the locking screw disengage the threads in the breech end; at the end of its travel the empty case is withdrawn from the breech.

VICKERS BREECH MECHANISM.

One of the latest patterns of breech mechanism for 12-inch guns of the Vickers type is illustrated in Fig. 24. One of its principal characteristics is the interrupted and stepped breech screw mounted on a forwardly-projecting spigot of a carrier hinged to one side of the gun.

The breech screw is arranged so that after it has been rotated sufficiently to disengage it from the breech opening of the gun it is swung directly out with the obturator without any longitudinal withdrawal. This not only reduces the time to operate the mechanism, but allows the actuating gear to be of the simplest type.

The actuating gear for the mechanism consists of a very powerful combination of levers in conjunction with a suitable

pinion and worm gear. The gear is arranged so that the maximum power is exerted when seating or unseating the obturator.

The obturation at the mouth of the chamber is effected by means of a modified De Bange obturator, protected by a front copper disk and front and rear steel rings. The obturator is secured to the breech screw in the usual manner by a mushroom-headed bolt and nut. This obturator bolt is pierced by a vent hole and chambered at its rear end for the reception of the firing primer, and on the same end of the bolt there is mounted the firing gear with its electric or percussion locks. The action of these locks is controlled by the movement of the breech mechanism when opening or closing.

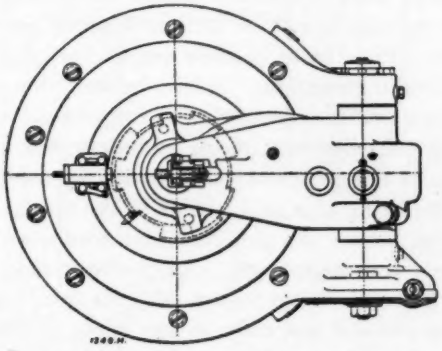


Fig. 24.—VICKERS 12-INCH BREECH MECHANISM.

Notwithstanding the great rapidity of movement when opening and closing such a breech mechanism absolute smoothness of action has been obtained, and the rebound and vibration is reduced to a minimum by the use of a small hydraulic buffer which absorbs the shock when opening, and a retaining latch which arrests any rebound of the breech mechanism when closing.

A retracting device is fitted which renders the firing gear absolutely safe before any movement has taken place in the opening of the breech.

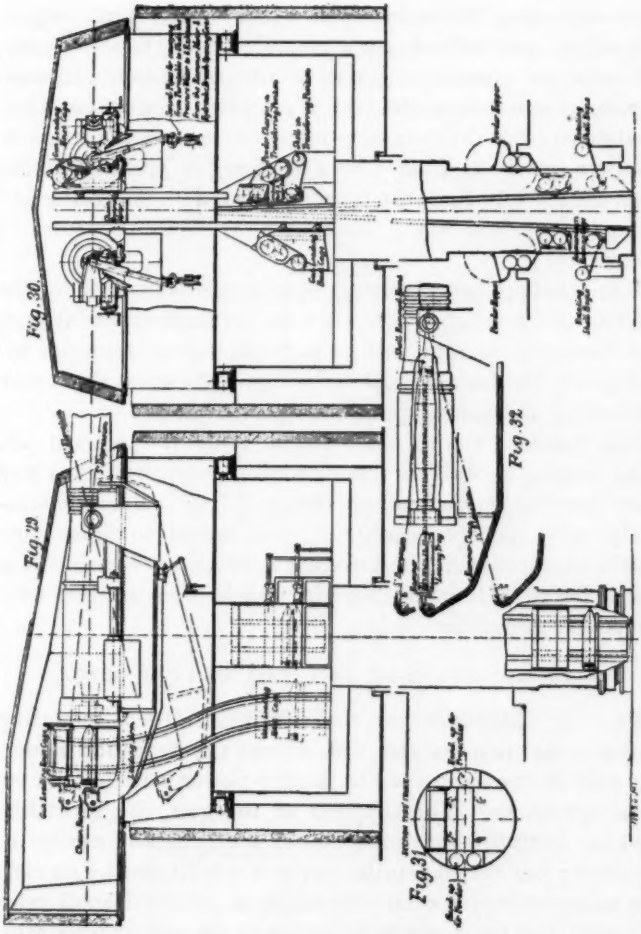
This breech mechanism is arranged so that it may be operated either by electric or hydraulic gear or by an auxiliary hand gear. Anti-friction rollers or ball bearings are employed in all the principal working parts. When operating this mechanism by hydraulic power the breech is opened in a little under four seconds and closed in a little over four seconds. The same operations can each be performed by hand power in six seconds.

HYDRAULIC MOUNTING FOR TWO 12–INCH GUNS.

Hydraulic power is most largely applied for the working of large guns at the present time, although electricity is being substituted in some instances. Before considering the merits of the two systems it may be well to describe generally their application. Figs. 29 to 31 show one of the latest examples of the hydraulic mounting. It will be seen that each gun is mounted in a cradle supported on a forged-steel slide, which also carries the necessary recoil cylinders and the run-out gear. One of the comparatively recent improvements introduced is the fitting also to this slide of the rammer for charging the gun so that the gun may be loaded even when it is elevated to a very high angle. As the hoist cage for serving the projectile and charge moves with the gun, loading operations may proceed not only at any angle of elevation, but also while the guns are being laid. This has greatly conduced to the high rate of fire and the improved accuracy now achieved.

The whole of the oscillating parts, viz: the gun, its cradle, slide and all other attachments, are so balanced on the trunnions that comparatively little power is required for elevating.

The turntable is a rigid structure built up of angle bars and plates, circular in plan, with the minimum of floor or platform area necessary. The outer wall of the turntable carries the armor shield, which sometimes weighs as much as 200 tons, and it rotates on rollers on a ring which is supported within the barbette armor. Suspended from the turn-



Figs. 29 to 31.—Hydraulic Mounting for Two 12-Inch Guns. Fig. 32.—Chain Rammer.

table is the working chamber and separate hoists for raising projectiles and explosive charges for each gun. Working chambers and hoists thus rotate with the turntable and gun. The training of the turret is effected by hydraulic motors, which, with their driving pinions and gear, are usually in duplicate and are attached to the side of the working chamber,

or in some cases to the turntable itself. These pinions gear with a rack secured to a machined face below the roller path, and turn the turret six degrees of arc per second. Electric motors are sometimes fitted as a provision against complete breakdown; but they are only intended to turn the turret at a slower speed—at from 1 to 2 degrees per second. Buffer stops are fitted to limit the training movement and bring the turret to a gradual stop. Locking bolts secure the turret in a seaway.

The training and elevating gear is all controlled by the movements of a handle, formed like a pistol, whose trigger fires the gun; in some cases hand wheels are employed for controlling the training and elevating. There are three such controlling and aiming positions in each turret.

The hoists are also actuated by hydraulic power; but these details, as well as the hydraulic recoil cylinders and other features for controlling the gun, I will illustrate separately, as it will be possible by this means to show more clearly the mechanical improvements which have enabled the rate of fire to be from six to eight rounds per 2 minutes with 850-pound projectiles.

ELECTRIC MOUNTING FOR TWO 10–INCH GUNS.

Fig. 33 illustrates an electrically-controlled mounting which, in its main features, differs from the hydraulic mounting only in the nature of the motive power used for the various operations. The supports of the gun, the turntable, working chamber and hoist trunks are generally similar in structure; but the hydraulic engine is substituted by an electric motor with a special controller, to which I shall refer presently, and frictional attachments to prevent undue straining. The motors for training the guns are arranged to turn the turret at a speed of three to four degrees of arc per second when the ship is inclined to eight degrees.

In the absence of hydraulic power the gun is returned to the firing position by springs or compressed-air cylinders, air pressure being always available from the torpedo departments.

In the mounting illustrated four sets of springs are provided, with sufficient power to hold the gun in the firing position at the maximum angle of elevation—35 degrees in this case. The elevating of the gun is also accomplished by electric

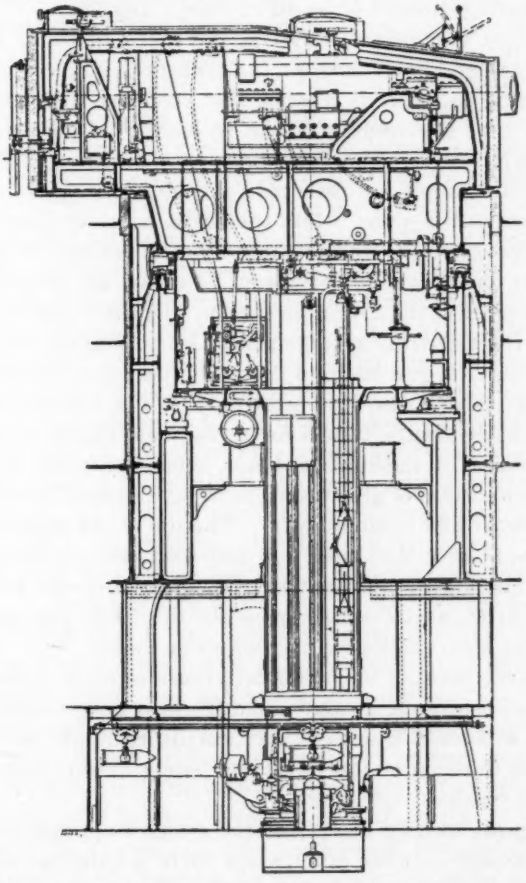


Fig. 33.—Electric Mounting for Two 10-Inch Guns.

motors with worm gearing. There is also in the working chamber hand gear, which is interlocked with the motors so that both cannot be in operation simultaneously. This hand

gear also rotates a worm actuating a pinion in gear with a curved tooth rack on the under side of the gun-slide frames. The ammunition elevators have electric and hand winches, and the rammer is also worked by an electric motor with worm gearing, instead of by an hydraulic engine.

Although the hydraulic system has many qualities which make it preferable for most of the operations connected with gun mountings, there is a belief that economy would accrue by the entire superseding of the hydraulic power plant by the electric system, as all ships must have electric generators, which could supply, at little cost, the additional units necessary for working guns. But the main consideration must ever be the reliability and adaptability of the system of power, and these qualities, it is considered by many, are present in a greater degree in the hydraulic system, when well constructed, than in the electrical arrangement. For elevating the gun, for running the gun back to the firing position, for working the rammers, and also for operating the lifts, the direct motion of the hydraulic ram is more easily applicable than the rotary motion of the electric motor, since the latter requires the multiplication of gear which, however efficient, is weighty, wears readily and is often noisy. The use of the Janney controller, to be described later, has overcome many of the objections of electricity, especially in its application to the training of the turret, as it renders possible the high precision of movement obtainable with the hydraulic motor.

From the point of view of safety there is much to be said in favor of hydraulic power. Damage is at once located and is easily repaired; whereas, short circuiting might seriously affect the working of the turret and fusing might cause ignition of explosives.

As regards weight, neither of the systems appears to have any advantage over the other when there is reckoned against the hydraulic power the water tanks, pipes and the reserve supply of water. But in the case of hydraulic mountings the weights are carried lower down in the ship than those of the electric system; this affects stability advantageously.

THE OPERATION OF LOADING AND FIRING THE GUN.

On this and the following page is a series of photographic views illustrating the loading of a gun on a modern ship.*

The first view shows the lower end of the hoist. The men have just run a shot into the cage. One man is seen working the levers which control an overhead railway traveler so arranged that by claws it lifts the shot from the bin and deposits it ultimately on to a tray fixed to the floor of the chamber. Around the trunk there is seen a toothed rack, which enables a second tray to travel around close to the trunk, so that even should the trunk be revolving with the whole mechanism of the gun, the projectile can be delivered by the traveling tray into the cage at any position.

The next illustration shows the magazine floor immediately over the projectile room, with the powder shelves to the left. Two quarter charges are being loaded into the hoist cage. Here also are trays on which the charges are placed in readiness to be transferred to the lifting cage. Neither shot nor powder can be passed into the trunk until the cage is in the proper position to receive it, as interlocking gear is provided. The cages for shot and powder are independent.

A view of the working chamber is next shown. The operator seen is at the levers for working the ammunition hoists. Dials are provided so that he can at once see the position of the cages. Speaking tubes are also fitted, as shown. The gear is all interlocked, and thus there is no possibility of premature hoisting or loading or of jamming.

The cages at the upper end of their travel come into the gun turret proper, and are guided along curved rails to bring the shot in direct line with the bore of the gun, irrespective of the angle of training or elevation of the gun. The first to arrive is the projectile. The mechanical rammer comes into contact with the base of the shot while the point is immediately opposite the breech. In the other view this rammer is seen driving the projectile into the open breech. Interlock-

* These photographs have not been reproduced.

ing gear is provided, so that the rammer cannot be operated unless the cage with its projectile is in the correct position behind the gun for the ramming home of the shot; nor can the cage be lowered until the rammer is housed. When the shot is out of the way the powder charges, which are contained in a receptacle at the side of the cage, are rolled out on to the shot tray, a portion of the charge at a time, and are similarly rammed into the gun chamber. A mechanically-moved tray is provided to protect the screw thread in the breech end of the gun.

QUICK-FIRING BREECH-LOADING GUNS.

The main principles of the mountings of quick-firing guns adopted for cruisers and for the secondary armament of battleships may be briefly indicated. For 7.5-inch guns, which fire a 200-pound projectile with a muzzle energy of 12,500 foot tons, two systems of mountings are made. In barbettes they are arranged singly or in pairs, in a manner analagous to that adopted for the 12-inch and 10-inch guns, as already described. Where they are located in casemates a central-pivot mounting is adopted. This central-pivot system of mounting is also used for the 6-inch and 4.7-inch guns.

The elevating gear is on the left side of the mounting and the training gear on the right side, so that separate gunners are required for these two operations. The "number" who lays the gun for elevation also controls the firing pistol. A separate gunner is required for the sight setting, sights being provided on both sides of the mounting, and suitably cross-connected, so that the two sights receive simultaneously the same adjustments for range and for deflection.

The gun and the cradle in which it recoils are carefully balanced on the trunnions, so that the operation of elevating the gun is effected with the least possible effort. For the purpose of training the gun the whole revolving mass, consisting of gun, cradle, top carriage and shield, etc., is balanced on a center-pivot ball bearing, which makes the train-

ing operation extremely simple and easy to control. With a 6-inch gun on such mounting ten hits per minute have been made in battle practice. The operating gear is so arranged that the gun may be laid and kept on its object, notwithstanding that the relative speed of the attacking ship and the target are sometimes very great. The speed and rolling of the ship has to be taken into account in determining the speed of the elevating gear. In order that these operations, particularly that of sight setting, may be effected with the utmost accuracy, anti-friction bearings and anti-backlash gear are introduced, whenever practicable, so as to reduce to a minimum any mechanical inaccuracies.

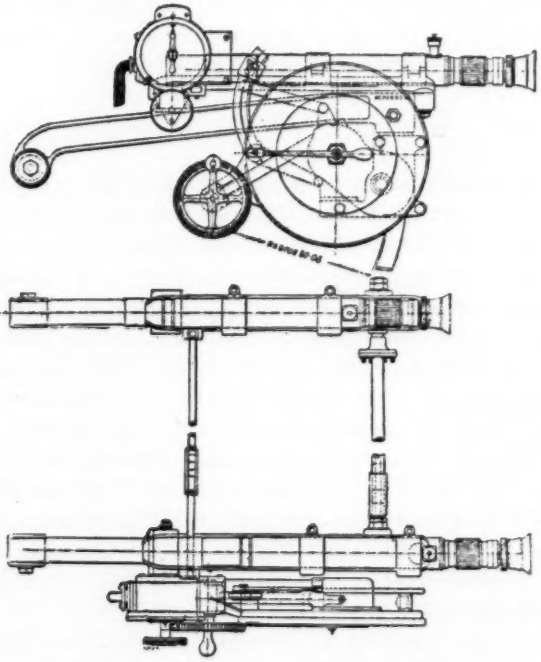
THE LATEST FORM OF GUN SIGHTS.

An adaptation of the telescope as a naval gun sight became imperative with high velocities, as it was not possible to get certainty and accuracy of fire at the greatly increased range with the open sights previously employed. The telescopic sight is now practically universally adopted, and a high degree of efficiency has been reached in its manufacture. Moreover, it can be illuminated for use at night. The next step was the duplication of the sights on a twin-gun mounting with cross-connection, so that they received simultaneously the same setting. Still further increased rapidity and accuracy of fire being aimed at, there has been a wide division of duties, so that each gunner may concentrate his attention upon one particular operation; thus, one man is now appointed to lay the gun for direction, another for elevation, while another operates the sights.

One of the latest forms of sight used is shown herewith. It combines with the sight the fire-control receiver. It is fitted with pointers in front of the range and deflection dials. These pointers are rotated by small step-by-step motors which are controlled from the range-finding station, and all that the sight setter has to do is to turn the elevating and deflection hand wheels until an index point on their respective dials comes opposite the pointer.

ACCURACY OF FIRE.

The result of the developments in the engineering side of artillery, in the optical accessories, and also—and in great measure—in the enthusiasm, training and skill of the gunners, is a marked improvement in the battle practice of the fleet. Now the target, which is towed at an unknown speed, course and range, is only 90 feet long by 30 feet high—less than one-fifth the length of a battleship. One of the latest



ELECTRIC "FOLLOW THE POINTER" SIGHT.

cruisers put on 18 hits out of 32 rounds from 12-inch guns, and another 15 hits out of 18 rounds. With the 6-inch gun 10 hits per minute have been made. Taking all guns in all British ships in commission, the percentage of hits to rounds

fired was 56 per cent. last year under the new and more severe firing conditions. It may be taken that efficiency in gun practice has more than trebled in ten years.

Range, yards.	12-inch gun (1864).		12-inch gun (1909).	
	Wrought-iron armor of 1864.	Krupp cemented armor of 1909.	Wrought-iron armor of 1864.	Krupp cemented armor of 1909.
	Inches.	*Inches.*	*Inches.*	*Inches.*
3,000	10	4.5	45	22
4,000	9	4.0	42	20
5,000	9	...	39	19
6,000	8	...	37	18
8,000			33	16.5
10,000			29	15
12,000			26	13
14,000			23	12
16,000	Outside range of 1864 gun.		21	11
18,000			19	10
20,000			18	9.5
22,000			17.5	9
24,000			17.5	9

This improvement in gunnery is, in part, due to the fact that the increase in the velocities and energies of modern guns has greatly flattened the trajectory, and consequently increased the length of the danger zone. The projectile for the modern gun has, at 3,500 yards' range, nearly ten times the energy of the projectile fired from the earlier guns at the same range.—" Page's Weekly."

UPKEEP OF THE PARSONS MARINE STEAM TURBINES OF THE U. S. S. *CHESTER*.

ADDITIONAL NOTES.

By Lieut. A. F. H. Yates, U. S. N.

Notes on this same subject were published in the last issue of the Journal, but the unexpected movements of the vessel prevented their completion. As was therein stated, the *Chester* left the navy yard after a partial examination of the internal conditions of her starboard low-pressure ahead and starboard astern turbine. Upon her return to the yard nine days later, this turbine was again opened for a complete examination and for such repairs as might be found necessary. Steps were taken to remove the upper half from the ship and take it to the machine shop. This was done in order to expedite the repair work as much as possible, and to leave the engine room unobstructed for other work. With the upper half of the casing out of the way, the spindle was next lifted and shored up above the lower half of the casing to a height of about six feet, slings being kept on the shafts in order that the spindle might be revolved in position to suit the demands of the work. This left sufficient head room below to permit of work going on in the lower half of the cylinder without interfering with that under way on the spindle. The turbine now being entirely open, a complete examination was made, and conditions in the lower half were nearly identical with those in the upper half. An examination of the low-pressure end showed that turbine to be in excellent condition and the blading without a blemish. An examination of the astern steam belt failed to disclose any provisions at this point for freeing it from water, and a complete survey of conditions in the whole astern turbine made it appear probable that the

blade stripping in this turbine was the combined result of a weak spot in the first row of fixed blades and an impulse of steam overcharged with moisture. No new developments were noticed as a result of the vessel's temporary absence from the yard and the attendant cruising. The blading stop pieces, which mark the end of each side of the cylinder rows of blades, appeared to be in good condition in this turbine, and in general there was no definite evidence that the stripping could have been caused by insufficient blade-tip clearance or uneven expansion in heating the turbine. Some rows of blades were affected by the loose fragments of stripped blades, but in most cases it was only considered necessary to file-finish the tips as neatly as possible without attempting to bend them. Blades which have been in service under varying temperature conditions must of necessity lose some of their elastic quality. In these rows, also, some of the lacing strips had numerous breaks, and it was planned to solder them at such points. The idea was not to expect to bring the strip back to its original strength, but to secure it together as well as possible, for new strips cannot be inserted without re-blading. In some of the later installations of the Parsons type of turbine the blades are put into place in blocks or sections, which has some advantage over the original method.

Work being under way on the starboard astern turbine, and delay being expected in securing the necessary material with which to complete it, it was planned to continue with the work on the high-pressure cruising steam glands and also to open and examine as many of the other turbines as the time would permit. Accordingly the opening of both of the cruising turbines and the port low-pressure and astern turbines was proceeded with. Opening of the two main high-pressure turbines was impracticable at the same time as that of the low-pressure turbines, so they were left out of consideration until other opportunity offered.

During the temporary absence of the *Chester* from the yard all turbines were in operation at one time or another, including the astern turbines, and while no really good oppor-

tunities were had for checking up the action of the astern turbines, yet the most was made of those that presented themselves. On one occasion when the port astern turbine was backed at half speed the revolutions per minute were but five less than what was considered normal. New and larger propellers being on at this time, this small difference was not considered as of any significance. Consequently, there was no reason for expecting to find any derangements in this turbine upon internal examination.

Lifting gear was rigged over the port low-pressure and astern turbine, and the work of lifting started. First, however, the nuts on the casing-joint stud bolts were removed. These for the greater part came loose with comparative ease, as in the case of the starboard turbine, but about ten or twelve worked hard and gave trouble, partially due to the cramped space in which the men had to operate their box wrenches and sledge hammers. The nuts once off, a strain was taken with the two five-ton falls making up the lifting gear. At this point real trouble was experienced, as the upper casing refused to break away from its joint. An examination showed that the stud bolts in the astern turbine casing joint were the ones giving the trouble. The strain was applied one way and another, and sledge-hammer blows and wedges employed, but to no avail. The main joint outside broke clear, and in places would open to $\frac{1}{8}$ or $\frac{1}{4}$ of an inch, but the astern turbine casing joint remained tight. The strain being on the upper half of the casing the jacking gear was used to see if the spindle would turn freely with that gear used lightly. It refused to turn at all; whereas, when the jacking gear had been used before the joint was to any extent broken, it turned with the greatest ease. This failure of the shaft to turn under the jacking gear indicated that, with a strain on the lifting gear, the lower half of the astern turbine cylinder was being sprung upwards sufficiently to cause the astern turbine blades on the lower half to bind on their tips against their opposite member.

The frozen stud bolts were given a kerosene bath and left to soak free. In this connection it should be remarked that

starting bolts are provided for the main joints of all turbines, but none are provided for the astern turbines. Owing to the relative inaccessibility of the latter, starting bolts could be of special value, but they should not be used, of course, unless the construction is such that the support of the lower half of the astern turbine casing will take all of the strain that results from their use. A renewed effort to lift the casing of this turbine was made on the following morning, but a deck beam supporting the after chain fall of the lifting gear started some of its rivets, so operations were stopped. Heat was then applied to the frozen stud bolts, being careful not to more than moderately heat the adjacent casing; but this failed to have any effect. Red lead and putty was then used, and a well built around each separate stud bolt in which was placed kerosene, and this was left there until the next morning. This gave the looked-for results, but it was after a day and a half of work that the casing was finally lifted.

The internal examination disclosed surprising conditions. In the upper half of the astern casing the first rows of blades in the first stage, a part of the rows in the third expansion, third stage, and the greater part of the rows in the fourth expansion, third stage, were found badly distorted, the worst place being in the inboard quadrant and in the neighborhood of the blading stop pieces on inboard side, the latter being deformed at the ends and sprung. The spindle blading showed similar conditions, but to a less extent. Re-blading being necessary in this astern turbine, the upper half of the casing was lifted out of the engine room and sent to the yard machine shop, and the lifting gear arranged for raising the spindle. The spindle was raised and blocked up, as in the case of the forward low-pressure turbine, and a complete examination of the remaining parts of this turbine was made. An examination of the lower half of the astern cylinder showed the first three rows of the first stage and the greater part of the third stage in need of re-blading. A general survey of the entire astern turbine showed some other rows in need of slight repair, such as it was proposed to make

in the case of the starboard astern turbine. A complete examination of the blading in the low-pressure turbine showed it to be in excellent condition. The ahead dummy was in good condition. In the low-pressure steam belt, however, there was found about half a bucket of small iron scale, lying over the drain opening, but apparently not clogging it. This scale might have come through the drain pipe from the port main high-pressure turbine, which discharges into this steam belt, or it might have been shaken loose in the low-pressure turbine when it was being opened, and dropped down; the latter being, perhaps, more likely.

The astern dummy is of the radial type. The strips on the spindle showed slight uniform wear. The dummy strips of the casing were in a similar condition, except that the after four strips in each case were worn down considerably. No repairs to these were considered necessary.

In the astern turbine blading no blades were stripped. Furthermore, the conditions as they appeared gave every evidence that the damage was due to insufficient blade-tip clearance. No drain was found in the astern steam belt, but water could not have been a cause for this particular damage. The condition of the inboard stop pieces and the blading in the adjoining quadrants showed conclusively that it was in these locations that the worst conditions of clearance existed. The original blade-tip clearances measured in this turbine (port low-pressure and astern) by the contractors were not satisfactory, it being found that those on the inboard side were much less than those on the outboard side, and even the mean clearance was slightly below the design figures. Nothing could well be done about the latter, and it was hoped that the removal of the eccentricity would completely remedy the situation. Another feature of the eccentricity was that it was a maximum at the astern end of the turbine and of a comparatively minor nature at the forward, or low-pressure ahead end. Accordingly the means taken to rectify matters was to correct the shaft alignment by babbiting the after main-journal bearing brass of the turbine off center, outboard.

This correction was in the right direction and, so far as it went, remedied conditions; but it is now believed that the blade-tip clearance was too scant, and mostly, too, on the inboard half of the turbines. It illustrates plainly how much care is necessary in construction and what extreme measures must be taken to avoid over-fine work.

A most unexpected incident was the discovery of appreciable wear on the forward face of the after main-journal brass, where the fillet of the shaft had worn against it. The writer had not considered such an occurrence possible without first experiencing the wrecking of the ahead dummy rings, it being his impression that the turbine could be adjusted for the full limit of dummy clearance consistent with the dummy piston-groove limits and still not encounter any such obstacle in shaft bearings, or elsewhere. At no time, by the micrometer, was this turbine ever allowed as great a clearance as one-half the limit which the width of the dummy-piston groove permitted.

For the purpose of securing blading material and calking-strip sections standard gauges are supplied, and the material in each case is supplied in lengths of about fifteen feet. These lengths are taken to the machine shop, where they are fed into special machines for cutting the blades and calking strips to proper lengths, and in addition the blades are stamped with calking grooves at their base, saw-cut to form the necessary recess for the lacing strip, and their ends champered off to a thin edge. In this connection it was found that the machines rigged for the purpose did not properly handle the material, and it was discovered that the reason therefor was the fact that they were made to handle material for blading the turbine of the battle ship *Florida*. The blade numbers in the case of the two ships were identical and, being of standard dimension, should have agreed; but a comparison showed that in reality the gauges differed slightly. The variation was sufficient to make it necessary to alter the gear for cutting the blades. While no great delay would result in such a case, yet it can be seen that considerable annoyance was experienced

by it, and there appears to be no good reason why strict standards should not be adhered to in such classes of work.

Another feature of this work which illustrates the extreme care necessary was as follows:

The grooves in spindle and cylinder into which the bases of the blades are calked are supposed to be $\frac{1}{4}$ inch deep. Consequently, in cutting a $\frac{1}{2}$-inch blade it should be cut $\frac{3}{4}$ inch length, to leave the $\frac{1}{2}$ inch over and above the flush surface of the spindle or the cylinder. These latter dimensions are, of course, the rough figures, and must be modified by a few thousandths of an inch to secure the proper tip clearance when the blades are in place. For the purpose of checking up and making the necessary allowance therefor the actual depth of the grooves in the spindle was measured carefully, and it was found that whereas they should have been exactly and uniformly of $\frac{1}{4}$ inch depth they were appreciably at variance, one groove from another. Some grooves were slightly less than $\frac{1}{4}$ inch deep, whereas others were considerably deeper, and some grooves were not uniform in depth. Although the differences were in the thousandths of an inch, yet it had to be considered in cutting the blades.

Another slight difficulty was experienced in repairing the slightly damaged sections of blading. This was in soldering the broken places in the lacing strip. Proper facilities are absolutely essential for this class of work. Silver solder is used, and a blow torch with an exceedingly small flame of intense heat must be used.

While the above work was in progress the two cruising turbines were being prepared for opening. Unfortunately but one of these could be lifted at a time, as the overhead trolley track made it necessary, in swinging the upper half of the casing of either one to one side, to leave the same suspended above the other turbine; consequently the intermediate-pressure cruising turbine was only prepared for opening. Its casing joint was broken so it would be free for lifting later on, and the high-pressure cruising turbine was entirely opened. The blading was found to be in excellent condition, but con-

siderably more rust and scale was found than was expected. The inside of the wrought-steel drum had scale up to a thickness of $\frac{1}{32}$ of an inch, and the cast-steel arms of the rotor had scale of $\frac{1}{16}$ of an inch thickness. The cast-iron cylinder had considerable rust on it, but it was loose, red rust. In a previous article I have stated that rust need not be expected, but this experience has of necessity altered my views. An important fact is that the rust and scale found in the low-pressure and astern turbines was very slight. The high-pressure cruising turbine is subject to much more severe conditions than the low, being run under high pressure and under a greater range, and frequently being run idle under very low temperature. It is believed that this is the reason for the excessive corrosion. Several patent paints are now understood to be on the market for use on the inside of turbines. Perhaps the most serious effect of corrosion was that found on the high-pressure cruising dummy piston. It was noticed on the face of the dummy piston where the end of the micrometer stem bears, and also on the faces of the raised portions of the serrations. It was particularly noticeable on the face of the dummy piston, and as a result of the corrosive effect the end of the micrometer stem had worn a groove in it several thousandths of an inch deep. Although a slight amount of corrosion at this point would have to be expected, yet the amount found was more than desirable. This face was accurately file-finished and scraped smooth. An odd condition in the case of corrosion on the faces of the raised serrations was that only the three forward ones showed bad corrosion, the others being gradually less affected from forward aft. Indeed, corrosion in general appeared worse in the forward end of the turbine than in the after end. The brass dummy strips in this turbine showed slight burring at their sharp edges, and this must have been caused by running under too scant a dummy clearance at some time or other. The same corrosive effect was found to exist in the low-pressure turbines, but it was too slight to be worthy of consideration.

On account of the conditions found in the dummy it was

considered advisable to check up the micrometer constant after assembling the turbine. This is done by fitting a keep ring in the lower half of the thrust block of the proper size for allowing spindle to come into zero clearance, at the same time hauling in on the upper half of the thrust sufficiently to allow the spindle to make a running test on zero clearance. Great care must be exercised in doing this. The expansion coupling would be disconnected, in the case of cruising turbines, and the turbine run idle under a very low steam pressure. Probably five or ten inches of vacuum would suffice. It is wisest, also, to keep a low vacuum in the condenser, say fifteen inches. These precautions are necessary, as the turbine is apt to run away and wreck itself, being run under no load. It is even well to have a man stationed by the soda cock in order to break the vacuum if an emergency arises. The turbine being run under these conditions, the grinding of the dummy-piston rings on the dummy piston may be heard. This should, of course, be conducted skillfully, so that this grinding noise is not excessive, for otherwise the dummy rings would be stripped and ruined. In case the noise is excessive it would mean that the trial should be stopped until a new keep ring capable of properly regulating the zero clearance is fitted. Once running in good shape, however, all that is necessary to secure the desired results is to get accurate readings of the dummy micrometer, as it can be seen at once that the difference between the micrometer readings with the end of the stem against the dummy-piston face and the stop piece, respectively, gives the desired constant. This constant should be stamped on the micrometer handle for ready reference and the old constant erased therefrom.

A few remarks on the subject of blade-tip clearance might be of interest. The clearance in an astern turbine cannot, of course, be compared equally with those in ahead turbines, because the astern turbine gets very much rougher use than the ahead turbine. It is seldom in a heated condition when steam is admitted to it, because it has been running idle in a vac-

uum when the main system is running ahead, and demands made upon it are usually sudden and without warning. Consequently much greater clearance in comparison is necessary over the blade tips. The ahead turbines, by reason of being always warm and in the best possible preparation, run with very low blade-tip clearance. The benefit derived is naturally greater where the pressures are highest, and in design an effort is made to favor the higher-pressure turbines in this regard. The size and shape of the revolving rotor and, possibly, the length of blades, are factors to be considered. In the *Chester's* installation the high-pressure cruising turbine had the smallest clearances, as would be the case in any similar installation. This clearance was .030 inch in the first stage, .035 inch in the second stage and .040 inch in the third stage. In the previous set of notes it was stated that as a result of 36,000 miles of steaming the high-pressure cruising main bearings had dropped .01 inch. This, it will be seen, had an appreciable effect in reducing the blade-tip clearance in the bottom part of this turbine, but the fact that the blades were found in fine condition upon examination shows plainly how close a margin can be allowed in a machine of this nature.

The work previously referred to on the steam glands of the high-pressure cruising turbine was undertaken at this time, and, in addition, a new floating snap ring was made and fitted, to take the place of one found broken. The material procured for making new gland-packing strips proved to be harder in quality than desired, but was successfully used. It should be of soft brass, as it has to be rolled into shape to fit the curve of the shaft.

An attempt was made to secure accurate measurements of the spindle for the purpose of determining whether or not it had undergone any changes in shape as a result of service. I have no figures on this, but consider it an interesting point and one well worth attention. I have suspected some warping in the case of the larger spindles, and have given some thought on the advisability of constructing the latter with a

through shaft instead of with dead ends at the point where the wheel between shaft and rotor drum is fitted.

Another point is the size of the casing-flange securing bolts. Those used on the flanges of the *Chester's* turbines appeared to be unnecessarily large.

In line of improvement it would be well to look to the design of a better steam gland than that at present employed. It would seem to be entirely practicable to devise a regular packing in lieu of the present mechanical arrangement, which is more or less unsatisfactory.

TEST OF MAIN CIRCULATING PUMP FOR U. S. S. *NORTH DAKOTA.*

The following is an extract from report of test made by Commander Emil Theiss, U. S. N., Inspector of Machinery at Fore River Shipbuilding Co.

The apparatus for the test consists of two steel tanks connected by a channelway 40 feet long, 11 feet wide, 2 feet 6 inches high, having a 10-foot weir at the end next to the pump.

The pump drew water from the suction tank, which holds nearly 10,000 gallons, and discharged through a 24-inch pipe into a 30-inch pipe of sheet steel, leading to a reservoir from which the water returned through the channelway back to the suction tank.

The head over weir was observed to thousandths of a foot by means of a hook gauge over a barrel 13.5 feet up stream, connected to the channelway through a 2-inch pipe.

The zero reading on the gauge was carefully measured at the beginning of each test, and readings corrected for same.

The specifications require the pump to be tested under a head of 25 feet, 5 feet of which will be suction, under such conditions to discharge 15,000 gallons per minute at 265 r.p.m.

On the first test of May 21st the suction head was set at 5 feet for the high-speed run, but as the amount of water in the tanks was kept constant throughout the test, the lift gradually decreased to 3 feet 3 inches. On the succeeding tests the lift was kept at 5 feet by regulating the water level at the beginning of each run.

The engine was piped to the 5-inch steam line of the exp. turbine and steam supplied entirely by the B. & W. boiler used for the same purpose.

The exhaust was led to the Wheeler condenser of the 27-inch exp. turbine and the condensed steam measured on the weighing tanks.

Readings were taken as follows:

Tanks weighed every six minutes and revolution counter read contemporaneously, thus giving the mean r.p.m. for each tank.

Hook gauge read every two minutes.

One or more sets of indicator cards were taken during each tank after the steam flow had settled, and the r.p.m. were read from the counter with the aid of a stop watch.

Other gauge readings were taken immediately after reading the revolution counter.

The following tests were made:

1st. With delivery head of 20.22 feet and suction head as given above, with back pressure on the engine from 21 inches to 26 inches of vacuum.

2d. With delivery head of 9.05 feet and suction head of 5 feet, with back pressure on the engine from 21 inches to 26 inches of vacuum.

3d. With delivery head of 9.05 feet and suction head of 5 feet, with the back pressure on the engine 5 pounds gauge.

The Francis formula, corrected for velocity of approach and end contraction, was used in figuring the weir discharge.

$$Q = 7.481 \times 60 \times 3.33\,(L - .2H)\,[(H + h)^{\frac{3}{2}} - h^{\frac{3}{2}}],$$

where Q is the discharge in gallons per minute;
L is the length of weir in feet;
H is the observed head over weir in feet;
h is the head due to velocity of approach, figured as follows:

$$\text{Velocity of approach} = \frac{\text{Approx. discharge in cu. ft. per sec.}}{\text{Area of cross section of stream in sq. ft.}} = \frac{3.33\,L\,H^{\frac{3}{2}}}{\text{Area in sq. ft.}};$$

$$\text{Velocity head} = \frac{(\text{Velocity of approach})^2}{64.4}.$$

H. Head on weir, feet.	Height of stream, feet.	Cross section of stream, sq. feet.	Approx. discharge, $3.33\,L\,H^{\frac{3}{2}}$, cubic feet per second.	Veloc. of approach, feet per second.	h. Head of approach, feet.	Corrected.		
						Head, $H+h$.	Length, $L-\frac{H}{5}$.	Discharge, gals. per min.
1.3	2.05	32.55	49.36	2.19	.075	1.375	9.74	23,190
1.1	1.85	20.35	38.42	1.89	.0554	1.1554	9.78	17,980
.9	1.65	18.15	28.43	1.567	.038	.938	9.81	13,210
.7	1.45	15.95	19.50	1.222	.0232	.7232	9.86	9,026
.5	1.25	13.75	11.77	.856	.0114	.5114	9.95	5,389
.3	1.05	11.55	5.47	.473	.0035	.3035	9.94	2,473

Curves of weir readings, I.H.P. and steam consumption were plotted against r.p.m. as abscissae, and fair curves drawn through. The remaining curves were derived from these by computing a number of points for even values of r.p.m.

In figuring the water H.P. the head due to friction was neglected, and the total head considered as the sum of the static head plus the velocity head, which was figured as follows:

$$\text{Velocity of disch.} = \frac{\text{Disch. in cu. ft. per second}}{\text{Area of delivery pipe in sq. ft.}};$$

$$\text{Velocity head} = \frac{(\text{Velocity of disch.})^2}{64.4};$$

$$\text{Water H.P.} = \frac{\text{Gals. per min.} \times \text{total head} \times 8.33}{33{,}000};$$

$$\text{Pump and eng. eff. } \% = 100\,\frac{\text{Water H.P.}}{\text{I.H.P.}};$$

$$\text{W.R.} = \frac{\text{Total steam consumption}}{\text{I.H.P.}}.$$

Engine, 11-inch — 22-inch × 10; diameter piston rod, $2\frac{1}{2}$ inches; crank shaft, 5 inches diameter; cranks at 90 degrees.

H.P. piston valve, $4\frac{1}{4}$ inches diameter; L.P. piston valve, $8\frac{1}{2}$ inches diameter; travel of valves, 3 inches.

Steam, 3 inches diameter; exhaust, 6 inches diameter.

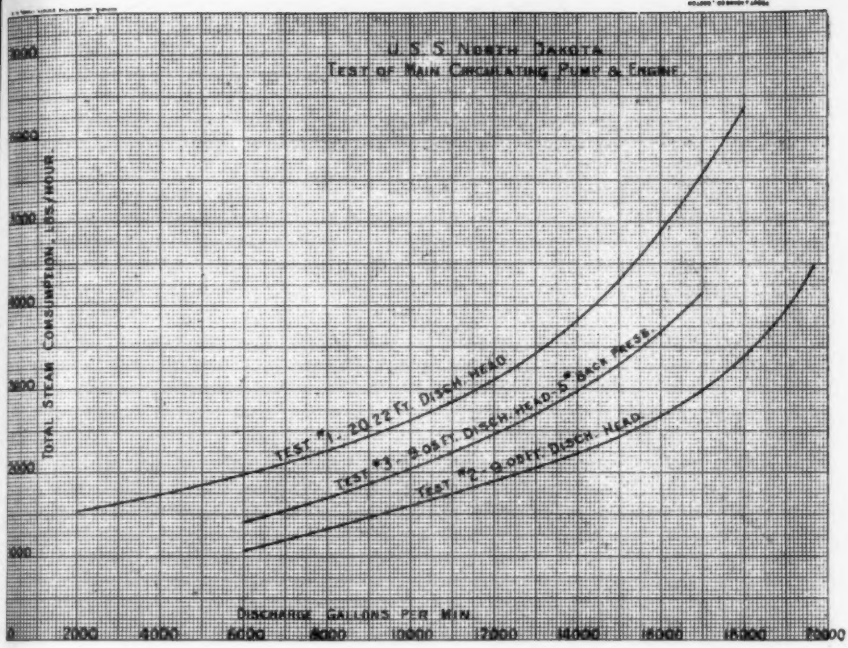
Pump, diameter suction, 24 inches; diameter discharge, 24 inches; impeller, 48 inches diameter; number of blades, 6.

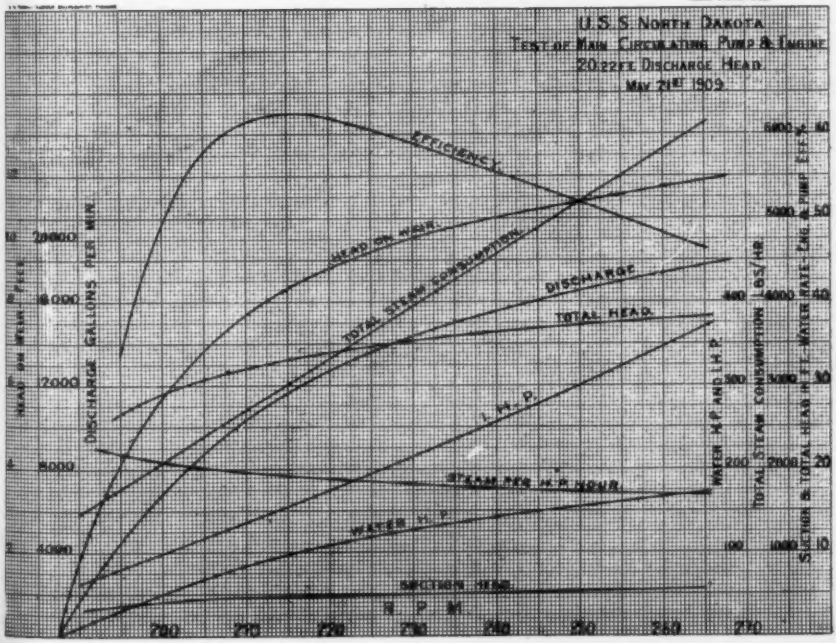
TEST OF MAIN CIRCULATING PUMP OF U. S. S. *NORTH DAKOTA.*

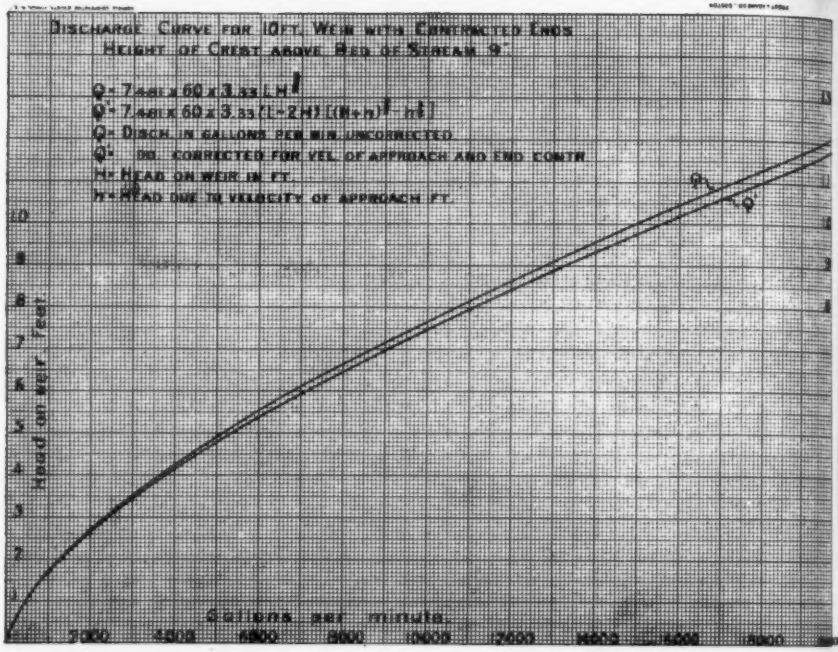
Test No. 1—May 21, 1909.

R.p.m.	Disch. gal.p.m.	Heads, in feet.				Water H.P.	I.H.P.	Eff. per ct.	Total st. con.	Water rate.
		Suct.	Disch.	Vel.	Total.					
195	5,000	3.55	20.22	2.84	26.21	26.9	80.0	33.6	1,750	21.85
210	10,400	4.45	20.22	7.39	32.06	84.1	137.0	61.4	2,700	19.7
225	13,600	4.80	20.22	9.65	34.67	119.0	196.0	60.8	3,650	18.65
240	15,500	5.00	20 22	11.00	36.22	141.9	257.0	55.2	4,570	17.78
255	16,900	5.30	20.22	12.00	37.52	160.0	322.5	49.65	5,500	17.0
265	17,650	5.50	20.22	12.52	38.24	172.6	370.0	46.6	6,150	16.6
					Test No. 2—May 24, 1909.					
155	6,800	5.00	9.05	4.85	18.85	32.4	52.0	62.2	1,175	22.6
170	10,770	5.00	9.05	7.64	21.69	59.0	83.0	71.1	1,700	20.5
185	14,140	5.00	9.05	10.04	24.09	86.0	118.0	72.9	2,270	19.25
200	16,800	5.00	9.05	11.92	25.97	110.0	157.5	70.0	2,900	18.4
215	18,560	5.00	9.05	13.18	27.23	127.7	202.0	63.2	3,650	18.05
230	19,650	5.00	9.05	13.95	28.00	139.0	250.0	55.5	4,485	17.9
245	20,280	5.00	9.05	14.4	28.45	145.5	300.0	48.5	5,400	18.0
					Test No. 3—May 25, 1909.					
155	...	5.00	9.05	...	...	...	80.0	...	1,525	29.35
170	...	5.00	9.05	...	...	...	137.0	...	2,175	26.2
185	...	5.00	9.05	...	...	...	196.0	...	3,030	25.7
200	...	5.00	9.05	...	...	...	257.0	...	4,000	25.4

Following curves were plotted from the data obtained :


U. S. S. North Dakota
Test of Main Circulating Pump & Engine
Total Steam Consumption - Lbs./Hour
Test #1 - 20.22 Ft. Disch. Head.
Test #3 - 3.08 Ft. Disch. Head. 5# Back Press.
Test #2 - 3.08 Ft. Disch. Head.
Discharge Gallons per Min.
0
2000
4000
6000
8000
10000
12000
14000
16000
18000
20000


U. S. S. North Dakota
Test of Main Circulating Pump & Engine
20.22 Ft. Discharge Head.
May 21st 1909
Efficiency
Head on Main
Total Steam Consumption
Discharge
Total Head
I. H. P.
Steam per H. P. Hour.
Water H. P.
Suction Head
Head on Main - Feet
Discharge Gallons per Min.
Water H.P. and I.H.P.
Total Steam Consumption - Lbs./Hr.
Steam & Total Head Ft. - Water Rate - Eng. & Pump Eff. %
R. P. M.
250
260
270


Discharge Curve for 10Ft. Weir with Contracted Ends
Height of Crest above Bed of Stream 9"
Q = 7.48 l x 60 x 3.33 L H^3/2
Q' = 7.48 l x 60 x 3.33 (L - 2H) ((H+h)^3/2 - h^3/2)
Q = Disch. in gallons per min. uncorrected.
Q' = do. corrected for vel. of approach and end contr.
H = Head on weir in ft.
h = Head due to velocity of approach ft.
Q'
Q
Head on Weir Feet
Gallons per minute.
7000
6800
6600
6400
6200
6000
5800
5600
5400
5200
5000
4800
4600
4400
4200
4000
3800
3600
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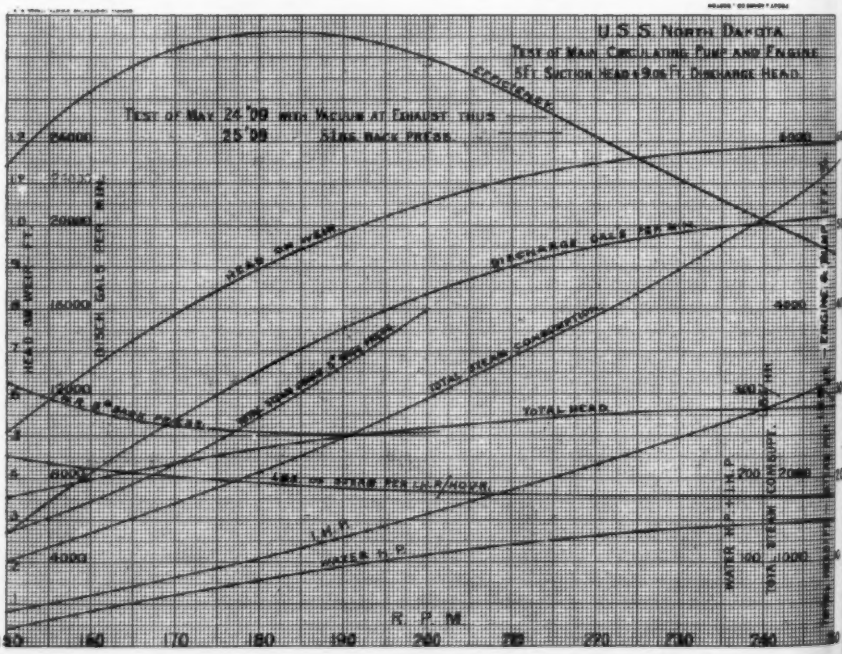
1800
1600
1400
1200
1000
800
600
400
200


U.S.S. North Dakota
Test of Main Circulating Pump and Engine
8Ft. Suction Head & 9.06 Ft. Discharge Head.
Test of May 24 '09 with Vacuum at Exhaust thus ——
25 '09 Zero back Press. ———
Efficiency
Head on Weir
Discharge Gals per min.
Total Steam Consumption
Total Head
Lbs. of Steam per I.H.P. per Hour
I.H.P.
Water H.P.
Head on Weir Ft.
Disch. Gals. per min.
Lbs. of Steam per I.H.P. per Hour
Water H.P. - I.H.P.
Total Steam Consumpt. per Hr.
Thermal Efficiency of Engine & Pump - I.H.P.
R. P. M.
150
160
170
180
190
200
210
220
230
240

TEST OF EVAPORATOR.

The following results and comments are taken from a report of test made with Bureau of Steam Engineering Standard Evaporator and Distillers at the U. S. Experimental Station, under supervision of Captain W. R. Worthington, U. S. N.

The rating marked on the evaporator tested is 3,250 gallons, and the two distillers are each rated as 1,500 gallons. The evaporator tested has fifty tubes each 2 inches O. D. by 52½ inches long, making the tube surface 114.53 square feet. The effective surface of the two heads is 9 square feet, as nearly as can be measured.

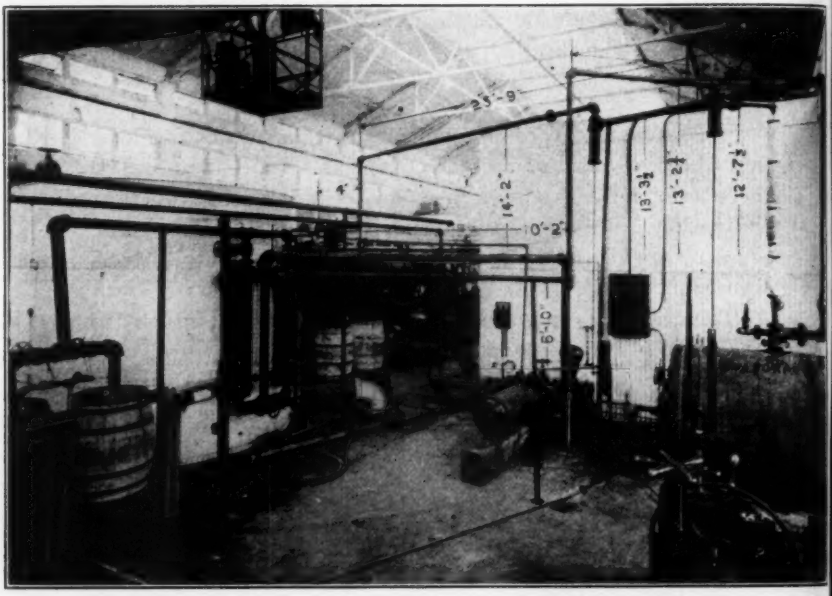
The evaporator was piped (as shown in the photograph) to the distillers and also to a condenser. The necessary weighing tanks were provided to collect the fresh water from the distillers, the salt water used for condensing the same, and the condensed steam from the circulating and feed pumps. All the cooling water through the condenser was measured by a water meter. The steam condensed in the coils was passed through an oil cooler to reduce its temperature below boiler and was then weighed.

The feed from the evaporator was taken from the discharge from the distiller, with the exception of tests 4, 5 and 8, when the condenser was used exclusively.

METHOD OF CONDUCTING THE TESTS.

All gauges, thermometers, water meter and scales were tested before and after the test and corrections made accordingly in the data obtained. Wherever a gauge was used a thermometer was inserted in the same space alongside as a check. It will be noted that the absolute pressures as entered in the

table do not in all cases agree with the temperatures, but, as the temperature could be read closer than the pressure, it was used in working up results. In taking the quality of the steam both a throttling and a separating calorimeter were used, and as their readings were very close together an average was taken as to the percentage of dryness. The water was tested with nitrate of silver each hour for salt, and found fresh.



ARRANGEMENT OF APPARATUS FOR EVAPORATOR TEST.

In addition a sample was also taken each hour for chemical test, and this also indicated absence of salt. The amount of water rejected after blow-down on account of salt is indicated in column 56, Table I.

The evaporator was scaled as carefully as possible, but could not be entirely freed of scale owing to the "distance bolts" blocking access to the spaces between some of the tubes. The evaporator was then filled with salt water to one-half glass

and started in the usual manner, and, as usual, considerable salt was found in the distilled water first coming over. Some of this was thought to be due to the intermittent action of the trap, which allowed the level of the water condensed in the coils to fluctuate a good deal. The use of the trap was then abandoned and the coils kept drained by hand, a small by-pass valve being fitting in addition to the full-sized one to facilitate adjusting the level of water in the coils which was afterwards held very uniform.

After a number of trials it was found best to carry the level of the salt water in the evaporator 2 inches below the top of the uppermost row of tubes, and this level was preserved with great uniformity throughout the test.

Instead of using a continuous blow it was thought more economical to blow down at intervals when the water was densest. The subsequent filling with cold water having a tendency to crack off the scale, it was assumed would prolong the time the evaporator could run without scaling.

The reducing valve on the steam pipe to the coils was found of some assistance in maintaining the pressure, but had to be assisted by hand regulation.

Owing to the difficulty in ascertaining whether the salt in the distilled water came from the distiller leaking or from the evaporator foaming, the pipe from the evaporator, which first rose only to the height of the top of the distiller and then ran horizontally, was altered to the lead shown in the photograph, that is, run up above the distiller, with a drop, at the bottom of which was a pet cock. The fact that the water drawn off here showed much more salt than was found in the distilled water suggested the idea of an extension of the pipe at this point and the fitting of a glass water gauge. It was found that considerable salt could be drawn off in this way. Later two other separators, made up of a short section of pipe and of ordinary pipe fittings, were placed as shown in the photograph, Fig. 1, and also other drip collectors with water gauges. The amount and density of the salt water collected in all of them was obtained and entered in Table IV. These drips

and glass gauges were found very convenient, as the quantity collecting in a given time showed whether the evaporator was foaming, and about how much, making it unnecessary to test the distilled water as often as would otherwise have been required.

It was observed that the amount of water collected in the various separators and drips varied greatly with the conditions. In Test No. 6 there was an apparent tendency to prime, and the separator nearest the evaporator collected 49.5 pounds of water, or about 1 per cent. of the water evaporated, the salinity of which was 485.7 grains per gallon, but no salt could be detected in the water delivered to the feed tanks.

Noting the apparent good effect of a separator on the line, it was thought well to try it under extreme conditions. Before starting Test No. 7, and while the 2-inch valve was open to the distiller, the 3-inch valve to the condenser was opened very rapidly, causing the evaporator to prime. The weight of water and powdered scale caught from No. 1 separator on the line to the distiller was 83 pounds, and from No. 3 separator on the line to the condenser was 104 pounds. While the water to the feed tanks indicated salt under this severe condition. In 15½ minutes no salt could be detected. No attempt was made to settle upon the exact type, dimensions or proportions of separator necessary to give the best results.

As still another test, the safety valve was made to lift causing the evaporator to foam badly, carrying powdered scale and salt water over into the separators and drips, and salt into the distiller in spite of the separators, the water becoming fresh in 7 minutes from the time the safety valve closed. Once this salt gets into the pipes a considerable time is required to wash it out, and a large waste of fresh water occurs. This could probably be obviated by making the evaporator shell strong enough to carry a pressure three or four times as great as the designed working pressure and in setting the safety valve at that high point to reduce the chances of its lifting.

On one occasion, when all conditions were apparently uniform, the water was observed to lift several inches in the glass

without any apparent cause. This may have been due to cracking off of pieces of scale.

In obtaining the data for Table I the following method was pursued in all cases: The evaporator was pumped up to a mark on the glass indicating a line 2 inches below the top of the top row of tubes. The discharge valve to the distiller or condenser (depending on which was used) was opened wide, and steam turned on the coils gradually, taking about ten minutes to reach the maximum pressure to be carried. When the distiller began to make fresh water (which varied from 9 to 18 minutes) the test was started. Readings of gauges, thermometers, calorimeters, etc., were taken every ten minutes and averaged for the entire run. Condensed steam from the coils and auxiliaries was weighed every ten minutes, while the distilled water was weighed at 5-minute intervals. The circulating water from the distillers was run into weighing tanks and weighed at intervals of about two minutes. All tests, with the exception of Nos. 5 and 7, were run for 4 hours, the above mentioned tests being stopped when the saturation reached such a point as to materially affect the output of the evaporator. At the end of the test the valve on the evaporator was immediately closed and the bottom blow opened. The evaporator was completely blown down and the discharge was weighed in tanks partly filled with cold water to prevent loss from reëvaporation. This operation complete, the bottom blow closed, the valve to distiller opened, the feed pump started and the evaporator pumped up to a level just above the top row of tubes and then drained down to the working level; this was done so as to crack off the scale that had formed on the top row of tubes. Steam was now turned on and the evaporator brought back to working condition, and distilling fresh water when the test was stopped.

During each run all conditions were maintained as uniform as possible, and a constant pressure was maintained in the coils.

Tests Nos. 1 and 2 were run at approximately rated capacity, the average being 3,353 gallons in 24 hours (allowing for the

time lost in blowing down) with 27.5 and 33.5 pounds gauge in the coil, the distiller alone being used.

Test No. 3 was also run with the distiller alone but at a high pressure in the coil, viz: 38 pounds gauge, giving 4,027 gallons of fresh water for 24 hours (allowing for blow-down).

It was noted after running Tests Nos. 1, 2 and 3 that an increase in pressure in the coils did not produce an equivalent increase in the water distilled, and after calculating the velocity of the steam in the 2-inch pipe to the distiller it was found that the velocity exceeded 16,000 feet per minute, as shown in Table IV. A condenser was then rigged up as a distiller, the air pump being cut out, and the circulating pump only being run, a 3-inch pipe conveying the steam from the evaporator to the distiller. Test No. 4 was now run, the pressure in the coil being 38.5 pounds gauge, the same as for Test No. 3, 5,186 gallons was distilled in 24 hours against 4,027 gallons (allowing for blow-down in both cases) using the 2-inch pipe, while the pressure in shell was reduced from 12.72 pounds to 7.07 pounds gauge, and the velocity in the pipe from 16,952 to 12,183 feet per minute, with a slight difference in the cost of evaporation in favor of Test No. 4, as shown by Table III.

Test No. 5 was run to find the maximum capacity of the 3-inch pipe, using the condenser as a distiller, as in Test No. 4. The pressure in the coils was carried at 52.5 pounds gauge, giving 12.07 pounds in the shell and distilling 6,607 gallons in 24 hours (allowing for blow-down), or more than double the rated capacity.

Having found that the 2-inch pipe to the distillers was too small for the capacity of the evaporator, in Test No. 6 it was thought advisable to run one distiller using the 2-inch pipe and find its maximum capacity. The left-hand distiller was cut off from the steam line and the output of the evaporator with 38.5 pounds gauge in the coil (the same as for Tests Nos. 3 and 4) was put through the right-hand distiller with the result that 3,431 gallons in 24 hours (allowing for blow-down) was distilled, more than double the rated capacity of the distiller, and more than the rated capacity of the evaporator.

The velocity in the piping in this case, was 13,906 feet per minute, the reduced velocity being due to the inability of the distiller to handle any more steam. During this test it will be noted from Table I, column 48, that the distilled water was returned to the feed tanks at 115.9 degrees Fahrenheit. The water could have been cooled much lower, but, owing to the lack of sufficient facilities for weighing the circulating water, the circulating pump had to be run at a reduced speed.

Test No. 7 was run to find the maximum capacity of the evaporator, using two distillers and the condenser as a distiller. 66.5 pounds gauge was carried on the coils, giving a shell pressure of 10.28 pounds gauge. In this case the distillers gave 2,990 gallons fresh water in 24 hours, approximately their rated capacity, and the condenser, 5,549 gallons in 24 hours (allowing for blow-down in both cases), or 8,539 gallons in 24 hours, 2.62 times the rated capacity of the evaporator.

Having noted that there was a slight difference in cost in favor of the test with the lower shell pressure, Test No. 8 was run, using the condenser and maintaining a vacuum in the same of 28 inches. It was first thought that by carrying 27.5 pounds gauge in the coil and 28 inches vacuum in the condenser a vacuum could be maintained in the shell, but it was found that the velocity of the steam in the 3-inch pipe would not exceed 15,516 feet per minute, and that the shell pressure was 1.18 pounds gauge. Under these conditions 4,744 gallons were distilled in 24 hours (allowing for blow-down) with a slight saving in cost of evaporation (Table III), but it was found that the extra steam required for circulating purposes to maintain this vacuum brought the cost up to slightly exceed that in Tests Nos. 1 and 2.

The average cost in steam condensed in the coils per pound of water distilled for the eight tests was 1.271 pounds dry steam (reduced to 44.74 pounds absolute, or about 30 pounds gauge in the coils, and taking feed water at 70 degrees Fahrenheit); while the average cost in pounds of steam to evaporate and distil (run circulating and feed pumps), was 1,447 pounds dry steam (when reduced to the same pressure).

In all cases, to obtain column 40, Table I, allowance was made for blow-down and returning evaporator to working condition after same, amounting in Tests 1 to 6 inclusive to 35 minutes for each blow, and in Tests 7 and 8 to 40 minutes for each blow.

It was noted in all tests that a slight variation in the water level had a noticeable effect on the pressure in the shell, a fall in the water level of a quarter inch causing the pressure to rise in some cases as much as 2 pounds, and a rise in the level a corresponding decrease in the pressure.

It was also noted that when the saturation had reached 3¼ thirty-seconds that the rate of evaporation began to fall off and that when the saturation had reached 4¼, or above, it fell very rapidly. In Test No. 6, when the saturation reached 4⅝ thirty-seconds, and in Test No. 7, with a final saturation of 4⅞ thirty-seconds, the evaporation fell off 10.02 per cent. and 12.77 per cent. respectively.

TABLES.

Table I gives the average of the recorded data and the steam used and water distilled reduced to a B.T.U. basis.

Table II records the heat balance made from Table I, dividing the B.T.U. there obtained by the pounds of water distilled, as found in column 39, Table I, thus reducing the B.T.U. expended to a basis of 1 pound of water distilled.

Table III records the cost of evaporation in terms of the steam condensed in the coils per pound of water distilled, and the cost of running the auxiliaries in pounds of steam per pound of water distilled.

Table IV records the density of the salt water in the shell, velocity of steam in the pipes, pounds of water extracted from the steam in the piping from evaporator to distiller by the separators, and the salinity of the same. The separators were numbered 1 and 2 on the line of the distiller and 3 and 4 on the line of the condenser, beginning with the one nearest the evaporator. Separators Nos. 2 and 4 consisted of only drop

pipes, as shown in photograph, and it is shown in the Table IV, by the small amount of moisture extracted, that they were not essential.

CONCLUSION.

It was found necessary, in order to obtain fresh water and maintain a satisfactory rate of evaporation, to insert a separator in the line to the distiller. This separator not only allowed of the making of fresh water, but when the evaporator primed it was found that all the scale and dirt from the evaporator carried over by the steam was collected here and drawn off, and that fresh water could be obtained in the tanks in a fraction of the time required without a separator.

The water level should be maintained at a constant point in the evaporator shell, for if allowed to fall materially the safety valve will lift, causing priming.

The steam should not be turned on the coils until the discharge valve to the distiller (and condenser, if in use) has been opened wide. It was found that if this was not the case it was almost an impossibility to open them afterward without causing priming. In starting Test No. 8 steam was turned on the coils with 27 inches vacuum in the shell and fresh water was obtained in 15 minutes—5 minutes from the time that water started to discharge from condenser.

The piping from the evaporator to the distiller for the 3,250-gallon evaporator should be increased from 2 inches to 3 inches, thus allowing an output of double the rated capacity of the evaporator and distiller in fresh water, as it was found that the capacity of the plant as designed and installed was limited by the size of this pipe and that the velocity of the steam would not much exceed 15,000 feet per minute.

That the total cost of evaporation and distilling in terms of steam used is affected very slightly by the pressure carried in the shell. While the cost in pounds of steam condensed in the coils is somewhat reduced by the lower shell pressures, this difference is made up by the extra cost in circulating water, when a condenser is used to maintain the lower shell pressure.

TABLE No. I.

DATA AND RESULTS OF TEST OF 3,250-GALLON EVAPORATOR.

No. of test.	Date of test, September, 1909.	Duration of test, in hours.	Barometer, corrected to 60 degrees Fahr.	B.T.U. supplied evaporator.																														
				B.T.U. supplied evaporator coils in steam while distilling.											B.T.U. supplied evaporator coils, in steam, to return evaporator to working condition after blow-down.									B.T.U. supplied evap'r in feed water while distill'g.					B.T.U. supl'd evap. in feed water, to ret. to working condition after blow.			
				Press.		Temp.									Press.		Temp.							Temp.											
				Gauge.	Absolute.	Inlet.	Discharge.	Per ct. moist. in steam.	Pounds saturated steam used.	Pounds moisture in steam.	Pounds dry steam condensed in coils.	B.T.U. supplied coils.	B.T.U. discharged from coil and returned to tank.	B.T.U. supplied to evaporator.	Gauge.	Absolute.	Inlet.	Discharge.	Per ct. moist. in steam.	Lbs. sat'd steam used.	Lbs. moist. in steam.	Lbs. dry steam condensed in coils.	B.T.U. supplied coils.	B.T.U. discha'd from coil and returned to tank.	B.T.U. supplied evaporator.	Sea water.	Feed water.	Pounds water fed to evaporator.	B.T.U. in feed water.	B.T.U. gained by taking feed from distiller circ. water disch.	Temp. feed.	Lbs. water fed to evapr'tor.	B.T.U. in feed water	Total B.T.U supplied evaporator.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35
1	9	4 1/6	30.01	27.5	42 24	270 0	252 6	1.47	6,269	92.15	6,176 85	7,213,738	1,387,957	5,825,781	19.5	34.24	257	212	1.42	220	3.12	216.88	252,347	39,666	212,681	72.5	107.9	5,338	405,154	188,431	75.0	11.54	49,737	6,493,353
2	10	4 1/6	29.77	33.5	48.13	277.2	255.7	1.15	6,376	73.32	6,302.68	7,370,146	1,431,858	5,938,288	31.5	46.13	275	214	1.90	235	4.47	230.53	269,848	42,841	227,007	72.0	110.0	5,440	424,320	206,176	72.5	11.41	46,324	6,635,939
3	13	4 1/6	30.08	38.5	53.27	285.2	264.9	2.31	7,533	174 01	7,358.99	8,646,679	1,761,969	6,884,710	34 0	48.77	280	225	2.36	390	9.20	380.80	446,796	75,426	371,370	73.7	111.7	6,466	515,340	245,061	75.0	11.59	49,953	7,821,373
4	14	4 1/2	30.07	38.5	53.27	285.6	261.6	2.07	9,873	204.37	9,668 63	11,355,530	2,276,714	9,078,816	38.5	53.27	285	227	2.26	489	11.05	477.95	561,484	95,551	465,933	74 8	74 8	8,300	356,070	...	75.0	11.75	50,642	9,951,461
5	15	3 1/6	30 05	52.5	67.26	300.5	277 1	2.63	9,905	260.50	9,644.50	11,389,165	2,440,592	8,948,573	47.5	62.26	295	245	2 64	722	19.06	702.94	828,803	154,291	674,512	75.4	75.4	8,254	359,049	...	76.0	11.87	52,346	10,034,480
6	16	4 1/6	29 90	38.5	53 19	285 0	264 8	2.38	6,595	156.96	6,438.04	7,565,322	1,541,911	6,023,411	34 5	49 19	280	240	2.25	407	9.15	397.85	466,682	84,900	381,782	75 1	109.0	5,534	426,118	187,049	76.0	12.49	55,081	6,886,392
7	17	2 3/4	29.91	66.5	81.20	312.9	284.5	3 139	9,206	288.14	8,917.86	10,581,192	2,337,864	8,243,328	55.0	69.70	302	269	2.85	871	24 82	846.18	1,000,239	207,385	792,854	75 0	111.6	7,943	632,263	289,920	75.0	13.10	56,461	9,724,906
8	20	4 3/4	30 14	27.5	42 30	270.5	233.6	1.88	8,857	166 51	8,690.48	10,159,510	1,790,000	8,369,510	28.5	43.30	272	246	1.88	596	11.20	584.80	683,928	127,961	555,967	73.4	73.4	7,717	320,253	...	74.0	12.26	51,615	9,297,345

TABLE No. I—Continued.

DATA AND RESULTS OF TEST OF 3,250-GALLON EVAPORATOR.

No. of test.	B.T.U. discharged from evaporator, lost in circulating water and returned to tanks.																					B.T.U. supplied circulating and feed pump.								
	B.T.U. in distilled water.													B.T.U. lost in blow-down.			B.T.U. lost overboard in obtaining fresh water.			B.T.U. lost overboard in circulating water.										
	Press. in shell.					Lbs. water in drip from separator.											Press. in shell.					Temp.			Pressure.		Temp.					
	Gauge.	Absolute.	Temperature in shell.	Pounds water distilled.	Equivalent gallons in 24 hours.	No. 1.	No. 2.	No 3.	No. 4.	Total drip.	B.T.U. in steam to distiller.	B.T.U. in drip.	Temp. discharge to feed tank.	B.T.U. in dist. water returned to tanks.	Initial temperature.	Pounds water discharged.	B.T.U. lost in blow-down.	Gauge.	Absolute.	Temperature.	Pounds discharged overboard.	B.T.U. lost overboard in obtaining fresh water.	Inlet.	Outlet.	Pounds circulating water.	B.T.U. lost overboard in circulating water.	Gauge.	Absolute.	Admission to pumps.	Water returned to feed tanks.	Per ct. moist. in steam.	Total lbs. saturated steam used.	B.T.U. in steam supplied pumps.	B.T.U. returned to feed tank.	B.T.U. absorbed by pumps.
0	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70
1	9.34	24.08	237.3	5,299	3,319.2	35.0	4.00	...	...	39.00	6,116,636	8,030	108.5	405,374	238	1,046	216,104	8.50	23.24	236.0	98.0 2 drip.	113,491	72.5	109.1	148,937	5,436,201	134.8	149.54	358.5	101.0	2.45	892	1,043,729	61,548	982,181
2	9.92	24.55	238.6	5,406	3,386.4	30.5	3.75	...	...	34.25	6,249,400	7,097	109.8	420,587	240	1,050	219,030	9.77	24.40	238.5	84.5 2½ drip.	98,034	72.0	110.6	143,980	5,543,230	124.0	138.63	352.1	102.0	1.91	862	1,010,936	60,340	950,596
3	12.72	27.49	245.4	6,429	4,027.2	33.0	4.00	...	...	37.00	7,437,196	7,922	124.8	596,611	247	1,057	227,995	13.82	28.59	248.0	102.0 2½ drip.	118,617	73.7	111.6	171,714	6,490,789	128.5	143.27	355.1	101.2	2.63	1,197	1,397,486	82,832	1,314,654
4	7.07	21.84	233.1	8,281c.	5,186.4	...	...	17	2	19.00	9,548,241	3,830	83.6	428,128	233	1,018	205,127	11.11	25.88	242.0	118.0 3 drip.	137,029	74.8	100.0	355,863	8,932,161	110.0	124.77	344.0	101.0	2.53	1,593	1,855,590	109,917	1,745,673
5	22.07	26.83	242.9	8,247c.	6,607.4	...	...	5	2	7.00	9,534,109	1,481	83.0	421,422	245	1,042	222,675	9.85	24.61	239.0	129.0 4½ drip.	149,852	75.4	105.0	301,683	8,899,649	125.5	140.26	353.4	101.5	3.02	1,504	1,750,054	104,528	1,645,526
6	14.04	28.73	247.9	5,482	3,431.7	49.5	2.00	...	...	51.50	6,345,799	11,155	155.9	679,220	249	1,072	233,374	15.51	30.20	251.0	192.0 2½ drip.	222,981	75.1	106.8	170,028	5,372,885	130.0	144.69	355.5	103.4	2.52	1,028	1,201,311	73,399	1,127,912
7	10.28	24.98	240.3	2,778d. 5,154c.	8,539.2	3.5	1.00	5	1	4.50 6.00	3,209,118 5,953,849	940 1,253	108.3 79.4	211,961 244,815	238	1,030	212,798	10.17	24.87	239.0	250.0 5 drip.	289,738	75.0 75.0	107.9 98.6	80,606 235,579	2,643,877 5,536,107	103.0	117.70	339.6	102.3	3.20	1,587	1,836,920	111,566	1,725,354
8	1.18	15.98	216.0	7,712c.	4,743.7	...	...	5	...	5.00	8,851,834	922	79.8	369,405	214	1,082	197,249	3.18	17.98	222.0	170.0 2 drip.	195,830	73.4	89.7	510,753	8,325,274	124.0	138.80	352.4	101.3	2.21	1,732	2,026,942	120,028	1,906,914

c = Condenser. d = Distiller.

TABLE No. II.—HEAT BALANCE.*

Item No.	B.T.U.	Test number. 1		2		3		4		5		6		7		8	
		B.T.U.	Per cent.	B.T.U.	Per cent.	B.T.U.	Per cent.	B.T.U.	Per cent.	B.T.U.	Per cent.	B.T.U.	Per cent.	B.T.U.	Per cent.	B.T.U.	Per cent.
1	Required to evaporate sea water at 32° F. and distil 1 pound of fresh water	1,691.77	...	1,687.31	...	1,719.75	...	1,712.27	...	1,743.59	...	1,772.07	...	1,778.50	...	1,717.09	...
2	Returned to feed tank in steam water from coils	269 42	15 92	272 79	16.17	285.80	16.62	286.47	16.73	314.65	18.05	296.75	16.74	320.88	18.04	248.70	14.48
3	Returned to ship's tanks in distilled water	76.50	4.52	77.80	4.61	92.80	5.40	51.70	3.02	51.10	2.93	123.90	6.99	57 58	3.24	47.90	2 79
4	Returned to feed tanks in steam water from circulating and feed pumps	11.62	0.69	11.16	0.66	12.88	0.75	13.28	0 78	12.67	0.73	13.39	0.76	14.06	0.79	15.56	0.91
5	Lost in drip from separator on pipe to distillers	1.52	0.09	1.31	0.08	1.24	0.07	0.46	0.03	0.18	0.01	2.03	0.11	0.28	0.02	0.12	0.01
6	Lost in circulating water from distillers	1,025.89	60.63	1,025.39	60.78	1,009.61	58.71	1,078.63	62.99	1,079.14	61.89	980.10	55.31	1,031.26	57.98	1,079.52	62.87
7	Extracted from distiller (circulating water) and returned as evaporator feed	35.56	2 10	38.14	2.26	38.12	2.22	...	...	...	...	34.12	1.93	36.55	2.06	...	...
8	Lost in blow-down	40.78	2.41	40.52	2 40	35.46	2.06	24.77	1.45	27.00	1.55	42.57	2.40	26.83	1.51	25.57	1.49
9	Lost (after blow-down from evaporator) to obtain fresh water	21.42	1.27	18.14	1.07	18.45	1.07	16.55	0 97	18.17	1.04	40.64	2.29	36.53	2.05	25.39	1.48
10	Lost in radiation from evaporator	7.38	0 44	12.83	0.76	4.61	0.27	6.91	0.40	15.32	0.88	13.40	0.76	7.23	0.40	6.68	0.39
11	Required to run circulating and feed pumps	185.33	10.96	175.84	10 42	204.49	11.88	210.80	12.30	199.53	11.44	205.75	11.61	217.51	12.23	247.27	14.40
12	Lost in radiation from distiller, piping, and otherwise unaccounted for	16 35	0 97	13.39	0.79	16.29	0 95	22.70	1.33	25.83	1.48	19.42	1.10	29.79	1.68	20.38	1.18
	Total	1,691.77	100.00	1,687.31	100 00	1,719.75	100.00	1,712.27	100.00	1,743 59	100.00	1,772.07	100.00	1778.50	100.00	1,717.09	100.00

* All data obtained from Table No. I.

Item No. 1 by the sum of Columns 13+24+30+34+68÷ Col. 39.
2 by the sum of Columns 14+25÷Column 39.
3 from Column 49÷Column 39.
4 from Column 69÷Column 39.
5 from Column 47÷Column 39.
6 from Column 61÷Column 39.
7 from Column 31÷Column 39.
8 from Column 52÷Column 39.
9 from Column 57÷Column 39.
10 by the sum of Cols. 46+47+52+57 subtracted from Col. 35÷Col. 39.
11 from Column 70÷Column 39.
12 by the sum of Items No. 2+3+4+5+6+7+8+9+10+11 subtracted from Item No. 1.

TABLE, No. III.

STEAM REQUIRED TO EVAPORATE 1 POUND OF WATER.

.....	Test number.							
	1	2	3	4	5	6	7	8
Pounds dry steam at 44.74 pounds pressure, absolute, required to evaporate 1 pound of water from 70 degrees Fahrenheit.....	1.258	1.262	1.265	1.254	1.290	1.309	1.304	1.226
Pounds dry steam (reduced to 44.74 pounds pressure absolute) required to run circulating and feed pumps to distil 1 pound of water.....	.158	.150	.175	.180	.170	.176	.186	.211
Total pounds of steam required to evaporate 1 pound of water from 70 degrees Fahrenheit and distil the same.....	1.416	1.412	1.440	1.434	1 460	1.485	1.490	1.437

TABLE, No. IV.

STEAM VELOCITY, SALINOMETER, ETC.

No. of test.	Ratio average pressure in coil to pressure in shell.	Salinometer Reading.		Velocity of steam in pipes to distiller, in feet per minute.		Moisture caught in separators and salinity of the same,							
						No. 1.		No. 2.		No. 3.		No. 4.	
		Start.	End.	2-inch pipe to distiller.	3-inch pipe to condenser.	Pounds moisture.	Salinity, grains per gallon.	Pounds moisture.	Salinity, grains per gallon.	Pounds moisture.	Salinity, grains per gallon.	Pounds moisture.	Salinity, grains per gallon.
1	.567	1-32	2 1-32	16,051	...	35.0	3.3	4.00	.25	...	...	...	...
2	.510	1-32	2-32	16,009	...	30.5	4.7	3.75	1.40	...	...	...	...
3	.516	1-32	3-32	16,952	...	33.0	12.6	4.00	.56	...	...	...	...
4	.409	1-32	4-32	...	12,183	...	...	...	...	17.0	32.3	2.00	2.30
5	.399	1-32	4-32	...	13,742	...	...	...	...	5.0	123.0	2.00	3.96
6	.540	1-32	2-32	13,906	...	49.5	485.7	2.00	1.40	...	...	...	...
7	307	1-32	4-32	15,966	13,469	3.5	960.0	1.00	5.10	5 0	80.0	1.00	3.60
8	.378	1-32	4-32	...	15,516	...	...	...	...	5.0	312.5	...	...

THE FORBES DISTILLER, ITS PRINCIPLE OF OPERATION, DESCRIPTION AND OBJECT.

The following description of a novel type of distilling apparatus abridged from a description given by the inventor, Mr. JOHN FORBES, gives a most interesting study of the question of marine distilling apparatus, and appears to show that very great improvement in economy may be possible by the successful application of the apparatus herein described.—H. C. D.

The apparatus on shipboard used for making fresh water from sea water ordinarily consists of two members: 1st, the evaporator, and, 2d, the distiller or condenser for killing the vapors generated in the evaporators. In the Forbes Distiller the two functions of evaporation and condensation take place in the same chamber, but the operation of evaporation and condensation are carried out on the two sides, outside and inside, of the tubes. It is therefore necessary to term it a distiller, as that word covers wholly the two operations of evaporation and condensation, which together constitute distillation.

The object of the Forbes Distiller is to make fresh water from sea water for the "make up" feed as well as for drinking, culinary and certain other purposes. The object is further to accomplish the above in an efficient and economical manner to a degree never before attempted successfully, and, moreover, by the use of an apparatus which is absolutely simple, not complicated, and which is quite as easy to operate as an ordinary single-effect or rudimentary evaporator.

The meritorious features of the Forbes Distiller and points of superiority claimed for it are as follows:

1st. Great efficiency and economy (1 pound of steam from

the boiler being capable of producing at least 7 pounds of distilled water).

2d. Continued and uniform efficiency and economy of operation.

3d. Simplicity of construction.

4th. Durability, and economy in material and labor for up-keep.

5th. Accessibility for cleaning and repairs.

6th. The positive non-salting of the evaporating surface.

7th. The positive deposition of lime in a predetermined chamber or unit of the heat exchange, which may be by-passed and cleaned.

8th. Ability to be continuously operated, it being unnecessary to shut down for cleaning, there being no salting or scaling of the heating surface proper and lime being disposed of as per Claim No. 7.

9th. No extraneous surface necessary, as the act of condensation is carried on by the evaporating surface in one and the same apparatus; moreover, no extra circulating or condensing water is necessary.

Before describing the principle of operation of this distiller it seems well to set forth briefly the several well-known physical facts upon which the various actions depend.

1st. That steam is condensed on coming in contact with a cooler surface than itself.

2d. That in the act of condensation *the latent heat* of vaporization, which forms the major portion of the total heat of any given body of steam, is given up.

3d. That the temperature of saturated steam is dependent upon its pressure, also that steam may have its temperature increased by mechanical compression.

4th. That counter-current or heat-transferring apparatus when properly designed will actually transfer practically the entire heat from one body of liquid to another.

5th. That most of the lime of sea water is deposited at 230 degrees Fahrenheit, and that all of this lime is dropped at or before 305 degrees Fahrenheit.

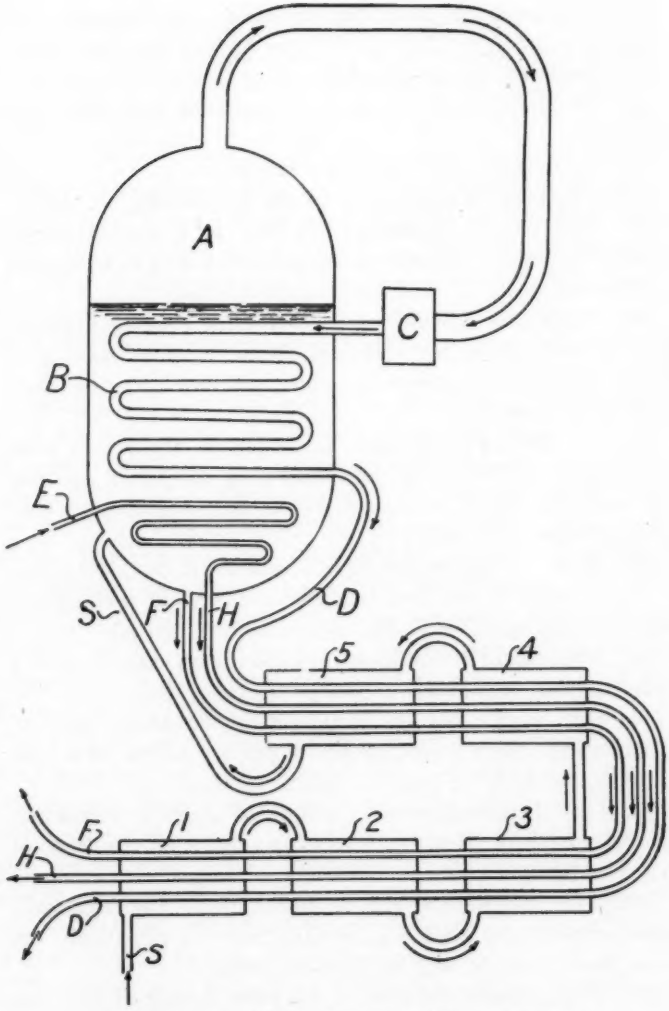


DIAGRAM OF FORBES DISTILLING APPARATUS.

6th. That there is practically no danger of salt deposition in a sea water saturation of 3/32.

Bearing in mind these physical facts above stated, the method of operation of the Forbes Distiller may be briefly stated thus:

1st. Covering only the acts of evaporation and condensation.—The withdrawal of the vapor from the evaporating or outside surface of the heating tubes at its normal saturated temperature, increasing this temperature slightly by mechanical compression, and placing it in its compressed and slightly hotter state on the inside surface of the tubes, thereby causing its condensation by reason of its coming in contact with tubes slightly cooler than itself, and at the same time causing it to give up its latent heat to the sea water to be evaporated on the evaporating or outside surface of the tubes.

2d. Covering the prevention of the salting of the evaporating tubes.—Flushing the evaporating chamber by the continuous inflow and outflow of sea water through the heat exchange in proper proportion to the amount of sea water being evaporated from it, thereby preventing the formation of salt by holding the saturation at or below 3/32, and withal not losing heat by the operation because of the high efficiency of the heat exchange.

3d. Covering the precipitation of lime in any desired zone.—The division of the heat exchange into several cells or units, in one of which the extreme thermal limits of the disposition of the lime are covered, thereby causing all the lime from the sea water on its way to the evaporating chamber to be deposited in one cell, and the adaptability by means of by-passing valves to deposit this lime in any predetermined one of the cells.

4th. Covering the retention and conservation of all heat within the apparatus, yet keeping up a heavy flush.—Causing the inflowing cold sea water on its way to the evaporating chamber to absorb, by means of the heat exchanges, practically all the heat of (1st) the distilled water being discharged by the apparatus; (2d) the heat from this slightly

concentrated sea water or brine on its way outward; (3d) the heat from the primary heating and radiation make-up coils.

In order to clearly show this method reference is made to the sketch in which *A* represents an evaporating chamber having within it the heating and condensing coil *B*. *C* represents a compressor having its suction side communicating with the evaporating chamber *A*, and its discharge side connected to the heating and condensing coils *B*. *E* represents the primary heating coil with its outlet *H*. *S* represents the inflowing sea-water supply to the evaporating chamber, and *F* represents the slightly denser sea-water or brine outlet.

Not considering the heat-exchanging element for the present, and taking a specific case and applying it to the distilling member, we shall assume that a certain quantity of sea water is within the evaporating chamber *A* and that the primary heating coil has been turned on and that the sea water is evaporating under a pressure of 150 pounds gauge pressure, with its temperature 365.9 degrees Fahrenheit. The compressor is now started and the vapor at 365.9 degrees Fahrenheit is withdrawn from the chamber *A* and compressed to 160.5 pounds gauge pressure with a temperature of 370.9 degrees Fahrenheit in coil *B*. The steam within the coil, therefore, is 5 degrees Fahrenheit hotter than the water in the evaporating chamber. The steam in this coil is therefore at once condensed and becomes distilled water, which is discharged at the outlet *D*, and in the act of condensation the latent heat of the compressed steam in the coil *B* is given up and absorbed by the sea water, to be boiled, into chamber *A*. The cycle of the latent heat is therefore through the compressor and coil and back into the water to be evaporated. This cycle may be performed a great number of times, and in this it is similar to a multiple-effect system, only the number of cycles is limited in the multiple-effect system by the number of effects, whereas in the Forbes system, it is unlimited, and continues until radiation has dissipated it. So far only the evaporating and condensing functions are explained. Taking up the heat exchanging function, which is represented in a purely diagrammatic

manner in sketch, 1, 2, 3, 4, 5, represent the several cells or units comprising the heat exchange. *S* represents the inflowing cold sea water which enters at the bottom of the cell 1 and has its outlet at the top, then entering the cell 2 at the bottom and leaving at the top, and so on, as shown by the arrows, until it finally emerges from the top of the cell 5 and enters the evaporating chamber of the distiller. *F* represents the hot brine from the evaporating chamber of the distiller. This brine flows down through a tube or a number of tubes in series through the five cells. The hot distilled water leaving the coil *B* likewise flows down through the five cells in a tube *H* or tubes. These three bodies of hot liquid brine, distilled water and trap water, leave the last cell of the heat exchange at only slightly higher temperature than the inflowing sea water, and in turn this inflowing sea water absorbs the heat of these three outbound bodies and leaves the heat exchange and enters the evaporating chamber at nearly the temperature of the sea water within the evaporating chamber.

It is therefore to be observed that but a very small portion of heat is leaving the entire apparatus and at the same time the evaporating chamber is being heavily and continually flushed, thereby preventing salting of its tubes.

Assuming that the sea water enters the 1st heat exchange from the sea at 60 degrees Fahrenheit and that it leaves the last or 5th heat exchange at 360 degrees Fahrenheit, then there is a difference of 300 degrees Fahrenheit occurring during transit, or 60 degrees Fahrenheit for each of the five heat exchanges.

As lime begins to deposit at 230 degrees Fahrenheit and finishes at or below 305 degrees Fahrenheit, the greater portion of the lime will be deposited in the 4th exchange. There is a sixth heat exchange, not shown in the diagram for the sake of clearness, but used in actual operation as a spare cell. This spare cell is used in turn by means of the by-pass valves when necessary to take the place of any one of the cells in the series, so that any particular cell may be cleaned or repaired

without interrupting the operation of the apparatus as a complete whole.

It has been found by the makers that in their heat-transferring apparatus, when operating on fresh water containing lime and other scale-forming elements, that these precipitates do not come down in the form of a hard scale on the transferring surfaces, but rather in the form of a sludge, which is easily gotten rid of by merely blowing down. This has been found to be a fact, resulting from the experience gained in the operation of many hundred heat-exchanging apparatuses designed and built by these manufacturers, and it is accounted for by them in the fact that no ebullition or evaporation takes place on this particular surface, and that there is not a great and active heat present—the degree of heat on the hotter side of the tube being not over 10 degrees Fahrenheit above that on the cooler side of the tube—and the process is therefore slow and not violent. It is therefore a very great probability that blowing down of the particular cell of the heat exchange in which the lime is designed to be precipitated will be sufficient to keep the apparatus free of lime.

Test of apparatus witnessed by Commander A. B. Zane, U. S. N., January 5, 1906, gave the following results.

Duration of test	1 hour.
Volts	110
Ampères	20
Revolutions per minute	133.9
Pressure in chamber	72.4
compression	81.5
live steam	98.1
Temperature, feed, degrees Fahrenheit	42.3
distilled, degrees Fahrenheit	70.9
brine, degrees Fahrenheit	70.5
trap discharge, degrees Fahrenheit	214.3
room, degrees Fahrenheit	75.8
Brake H.P. of motor for compressor	2.07
Live steam used, pounds	31.8
Water distilled, pounds	300.7
Steam for driving motor, pounds	68.2
Total steam used, pounds	100
Water distilled per pound of steam used, pounds	3

PROPOSED DISTILLER FORBES SYSTEM.

Capacity, 5,000 gallons per 24 hours.

Base pressure, 150 pounds gauge = 165 pounds absolute = 365.9 degrees Fahrenheit.

Compression, 10.5 pounds per square inch = 5 degrees Fahrenheit.

Temperature of feed, 60 degrees Fahrenheit (assumed).

Temperature of brine blow off, 67 degrees Fahrenheit (*i.e.* the heat exchanges are expected to work at a temperature difference of 7 degrees Fahrenheit).

Steam supplied by heating coil, 175 pounds per square inch gauge.

Ratio of brine blow off to distilled water, 1 :2 (*i.e.* the water in the evaporating element will be carried at a saturation of 3/32, which is considered very safe).

Radiation loss, 3 B.T.U. per square feet per degree difference of temperature per hour, which is perhaps excessive, but it is certainly on the safe side. 1.5 B.T.U. can readily be realized with proper lagging).

Heat transferred through tubes and coils, 350 B.T.U. per square foot per degree difference in temperature per hour. This is most conservative.

The Heating Surface of the Evaporating-Condensing Element.—The best authorities agree that for transferrence of heat between liquids and steam and liquids and liquids, the median line of the tube or other separating partition is to be taken as the surface to be figured on, and not the external or internal surface.

In the evaporator we have the heat of $208 \times 8.33 = 1{,}740$ pounds of steam at 160.5 pounds gauge pressure plus some heat as superheat; but as the higher temperature of this heat of superheat will carry it through the tube surface over and above the latent heat it is not necessary to figure on any surface for it, so we have:

Latent heat of 1 pound of steam at 175.5 pounds absolute pres-

sure=851.8 B.T.U., and at 350 B.T.U. per square foot per degree difference per hour we have $\frac{851.8 \times 1,740}{350 \times 5} = 848$ square feet.

This surface is to be in tubes and measured on the median line.

This surface will all be contained in a shell having not over 733 square feet external surface.

Size of Compressor Cylinder.—Steam at 150 pounds gauge has a volume of 2.751 cubic feet per pound, and for 5,000 gallons per 24 hours we have 208 gallons per hour, which is $2.751 \times 208 \times 8.33 = 4,760$ cubic feet of steam per hour.

If we use a double-acting single-cylinder compressor operating at 100 r.p.m. the cylinder will have a volume of

$$\frac{4,760 \times 1,728}{2 \times 100 \times 60} = 687 \text{ cubic inches.}$$

This size, however, includes no allowance for leakage, clearance, etc., and the superheating of the steam on coming in contact with the hot walls of the cylinder from which the compressed charge has just passed. By experiment this amount has been found to be 25 per cent. We shall, however, allow 30 per cent., and if the cylinder is found too large the speed of the compressor may be reduced a trifle. A 10-inch cylinder with 12-inch stroke will meet the requirements.

Radiation Loss will be $733 \times 3 \ (365.9 - 60 = 305.9) = 67,300$ B.T.U. per hour.

Loss of Heat in Water Leaving Apparatus.—In addition to the loss by radiation there will also be a loss of 7 B.T.U. for each pound of water leaving the apparatus, which will amount to 7,500 gallons per 24 hours, or $7,500 \times 8.33 = 2,600$ pounds per hour, or a loss of $2,600 \times 7 = 18,200$ B.T.U. per hour.

These losses, therefore, at $18,200 + 67,300 = 85,500$ B.T.U. per hour, which must be added to the apparatus to keep up the heat.

Part of this loss is supplied by the heat added to the steam

during compression, which is equal to the work done in the cylinder. We must ascertain this before we can tell how much surface we must add in the shape of a heating coil. The horsepower required to operate a 10-inch × 12-inch compressor, 100 r.p.m. against a pressure of 10.5 pounds pressure per square inch is, since $\frac{PLAN}{33{,}000} = \text{H.P.}$,

$$\frac{10.5 \times 1 \times 78.5 \times 200}{33{,}000} = 5.00 \text{ H.P.} = 12{,}700 \text{ B.T.U. per hour.}$$

Heating Coil.—The amount of heat to be added by the coil therefore is 85,500 — 12,700 = 72,800 B.T.U. per hour.

Assuming 175 pounds gauge pressure in the coil, the temperature difference between it and the evaporating chamber will be 377.4 — 365.9 = 11.5 degrees Fahrenheit.

As we have the same conditions in the coil as in the evaporating-condensing surface, if the air is properly taken off and the water of condensation properly trapped away we shall realize 350 B.T.U. per square foot per degree difference of temperature per hour, and we have $\frac{72{,}800}{350 \times 11.5} = 18.1$ square feet of surface coil.

We shall, however, add 50 per cent. to this coil to guard against possible drop in the pressure.

Feed Pump.—A single-acting feed pump, driven from the compressor crosshead direct, will have a stroke of 12 inches and capacity of about 439 gallons per hour; the power required to drive this pump being about .8 H.P.

Heat Exchange Surface.—This surface is divided into three parts, the incoming sea water bathing the exterior surface of all the parts, and the outgoing distilled water, brine and trappage from the heating coil bathing the interior surfaces. The surfaces, then, for these three elements are:—*Distilled-Water Exchange,* $\frac{208 \times 8.33 \times (365.9 - 67 = 298.9)}{350 \times 7} =$ 211.3 square feet, or say 212 square feet. *Brine Exchange:* As there will be one-half as much brine as distilled water we

take one-half the above, or 106 square feet. *Heating Coil Exchange:* Heat needed per hour, 72,800 B.T.U.

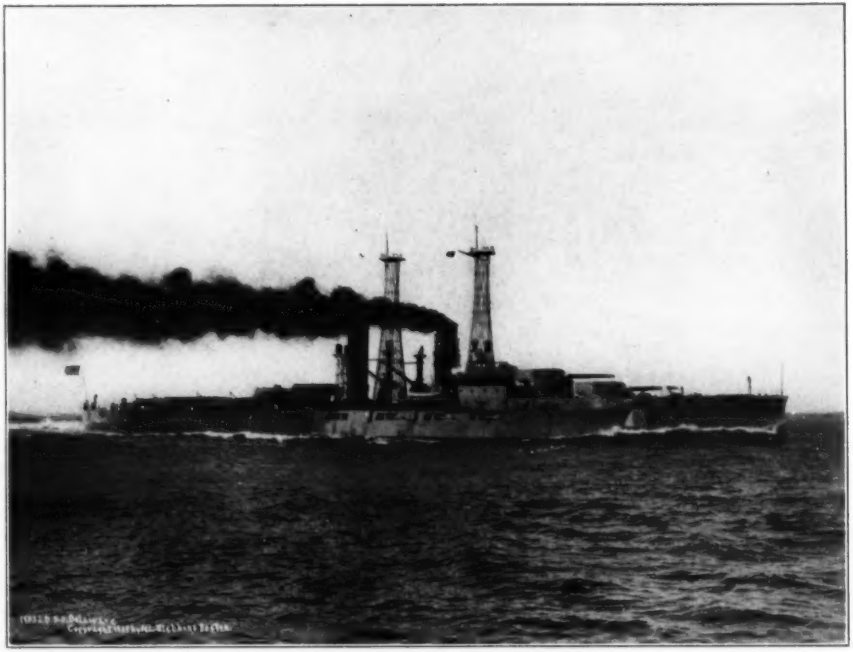
Heat in one pound of steam at 175 pounds gauge pressure above 32 degrees Fahrenheit = 1,197.1 B.T.U. Heat in one pound of water at 67 degrees Fahrenheit above 32 degrees Fahrenheit = 35.1 B.T.U. Then the available heat per pound weight of steam is 1,162 B.T.U. Then the steam required will be $\frac{72,800}{1,162} = 62.6$ pounds of steam.

It will be safer, however, to count on 50 per cent. excess of this amount, on account of leaks which must be made up by the coil, or 93.9 pounds of steam per hour, then

$$\frac{93.9 \times (365.9 - 67 = 298.9)}{350 \times 7} = 11.4 \text{ square feet.}$$

The total power required to operate the apparatus will be: Compressor, 5 H.P.: Pump, .8 H.P.: Total, 5.8 H.P.: and, assuming a motor efficiency of 90 per cent., a compressor efficiency of 80 per cent., and a pump efficiency of 60 per cent., it will require about 5.84 kw. delivered to the motor, which, with 31 pounds of steam per kw. hour and 8 pounds of steam per pound of coal, will give 22.63 pounds of coal equivalent of power per hour. The steam condensed in the heating coil amounts to about 62.6 pounds, or 7.82 pounds of coal, or a total of 30.45 pounds of coal per hour.

This gives $\frac{208 \times 8.35}{30.45} = 56.3$ pounds of water distilled per pound of coal.



U. S. S. "DELAWARE" ON STANDARDIZATION TRIALS.

U. S. BATTLESHIP *DELAWARE.*

By Capt. R. T. Hall, U. S. N.

DESCRIPTION OF MACHINERY—OFFICIAL TRIALS.

The battleship *Delaware*, built by the Newport News Shipbuilding and Dry Dock Co., Newport News, Va., is one of two sister ships, the other being the *North Dakota*, built by the Fore River Shipbuilding Co., of Quincy, Mass. The contract for this vessel was signed August 6, 1907, the price being $3,987,000, which does not include the armor and armor bolts (exclusive of protective deck), ordnance and ordnance outfit and certain articles supplied by the Government. The contract time for completion was thirty-six months.

The main engines were required to develop twenty-six thousand five hundred horsepower when working one hundred and thirty revolutions per minute, with a steam pressure of two hundred and sixty-five pounds at the high-pressure cylinder.

The guaranteed speed of the ship was twenty-one knots per hour for four hours.

PRINCIPAL DIMENSIONS OF HULL.

Length on normal load-water line, feet and inches,	510–00
between perpendiculars, feet and inches,	510–00
over all, feet and inches,	518–09
Breadth, molded, feet and inches,	84–10¼
extreme, to outside plating,	85–02⅝
Trial displacement, tons,	20,000
Mean draught to bottom of keel, feet and inches,	26–10¾
Displacement per inch immersion, at normal draught, tons,	71.6
Capacity of engine-room feed tanks, tons,	39
reserve feed-water compartments, tons,	221.4

MAIN ENGINES.

There are two propelling engines, right and left, outboard turning when going ahead, and placed abreast in watertight compartments, separated by a middle-line bulkhead.

The engines are of the vertical, inverted-cylinder, direct-acting, four-cylinder, triple-expansion type. The order of the cylinders, beginning forward, is forward low pressure, high pressure, intermediate and after low pressure. The cranks of the two forward engines are opposite, as are the cranks of the two after engines, the first pair being set at right angles with the second. The sequence of the cranks, therefore, is high pressure, intermediate pressure, forward low pressure and after low pressure.

The framing of the engines consists of forged columns trussed by forged-steel stays.

The engine bedplates are of cast steel, in three sections, supported on the keelson plates, and have seatings for the main bearings, columns and the turning engines.

The cylinders and valve chests are of the best quality of cast iron, fitted with working liners of close-grained cast iron as hard as can be properly worked. All cylinders are steam jacketed around the working liners and the intermediate and low-pressure cylinders at both ends. The steam is supplied at boiler pressure to the high-pressure cylinder jackets and through reducing valves for the intermediate and low-pressure jackets, successively, each having its respective drain through a trap to the feed tanks.

The high-pressure and the intermediate-pressure cylinders touch at finished surfaces so disposed as to offer constraint against athwartship motion only.

On the casings flanges are cast for securing lagging. There are facings for the crosshead guides and other fittings, and there are brackets for the supporting columns and the tie rods.

The high-pressure piston is made of cast iron, the intermediate and the low-pressure pistons of cast steel, and all followers are of class "A" cast steel.

The high and intermediate-pressure piston rings are solid,

the low-pressure rings are cut obliquely and fitted with a brass tongue piece and lug, over which is fitted a limiting piece. All the rings are set out with "C" springs, each having equal tension.

Each piston rod is tapered to fit the piston and it is secured by a nut having a locking plate. The lower end is fitted to a forged-steel crosshead to which is bolted a cast-steel slipper. This slipper is lined with white metal and works in a single guide bolted to the cylinder facings at the upper end and to a cast-steel guide of "I" section at the lower end; this girder is supported by the inboard engine columns.

The go-ahead guides are iron castings and are made hollow to contain a water jacket; the backing guides are of cast-steel and bolted to the flanges on the go-ahead guides.

The connecting rods are high-grade steel forgings, forked at the top to span the crosshead, and are "T" headed at the bottom.

The crank shafts are in two sections, each carrying two cranks, coupling disk and raised seatings for the eccentrics. The latter are made in two parts, bolted firmly in place and secured by a key with adjusting pieces on either side.

The forward section of the crank shaft operates a shaft bilge pump.

The main valves are of the piston type, one for the high-pressure and two for the intermediate and low-pressure cylinders.

The heads are of cast steel with lap-welded steel-pipe distance pieces riveted on; the packing rings are of cast iron turned larger than the bore of the valve chest, cut obliquely and bolted together to allow contraction but not expansion.

The valve stems carry balance pistons working in cylinders on the upper valve-chest cover; the tops of the high and of intermediate-pressure balance pistons are connected to the low-pressure receiver and the tops of the low-pressure balance pistons are connected to the main exhaust pipe.

The bottom of the high-pressure valve stem carries composition gibs working in brackets secured to the lower valve-

chest covers. The intermediate and the low-pressure valve stems are worked by a crosshead similarly guided.

The valve gear is of the Stephenson type, with double-bar links and an independent linking arrangement on the reversing shaft.

REVERSING GEAR.

The reversing gear for each engine consists of a steam cylinder and an oil-controlling cylinder bolted to the high-pressure cylinder. The piston rod of the reversing gear, which is common to both cylinders, acts directly on the arm, keyed to the reversing shaft. The piston rod passes through the controlling cylinder with a uniform diameter.

The valve of the steam cylinder is of the piston pattern, of composition, working in composition-lined valve chest.

There is a by-pass valve on the oil cylinder, worked by a continuation of the stem of the steam-piston valve. These valves are worked by a floating lever, the primary motion being derived from the hand lever on the working platform and the secondary motion from the reversing shaft, all parts being so adjusted that the reversing shaft follows the motion of the hand lever and is firmly held when stopped. There is a stopcock in the by-pass pipe of the oil cylinder, and a pump for reversing by hand is connected to the oil cylinder with its lever convenient to the working platform. The by-pass pipes are connected to the valve box of the hand pump in such a way as to leave the hand arrangement always in gear. The piston of the oil cylinder is packed by two cup leathers.

REVERSING SHAFT.

There is one reversing shaft for each engine, with an axial hole through it. It has arms for the reversing engine and for each link. Each reversing arm for the links is made with a slot fitted with a block, to which the extension links are attached. Each block is adjustable in the slot of its arm by a screw and hand wheel of an approved hand-locking device and is fitted with a suitable index. The slots in these arms are so arranged that the links may be thrown into full back-

ward gear, irrespective of the position of the block in the slot; and the length of the slots is such that cut-off may be varied from about 0.5 to 0.83 of the stroke.

TURNING ENGINE AND GEAR.

There is installed in each engine room a double engine for turning the main engine with steam of one hundred pounds pressure. This engine drives, by worm gearing, a second worm, which may be made, at will, to mesh with a worm wheel fitted on the crank shaft.

The turning engines have piston valves, and are made reversible by means of Stephenson links, reversed by hand levers.

Each turning-engine shaft is fitted for turning by hand.

WATER SERVICE.

Water service is provided for each engine by a $3\frac{1}{2}$-inch pipe from the discharge side of the main circulating pump. This pipe has suitable branches to the various parts of the engine; the discharge is returned to the suction side of the same pump by suitable pipe connections.

OIL SERVICE.

The oil service is supplied with oil from a gravity tank fed by a small steam pump with overflow pipe to the pump suction. The piping arrangement is such that oil is distributed to all the oil boxes, which have adjusting valves and wick feed to pipes leading to all bearings except those served by the forced-lubrication system.

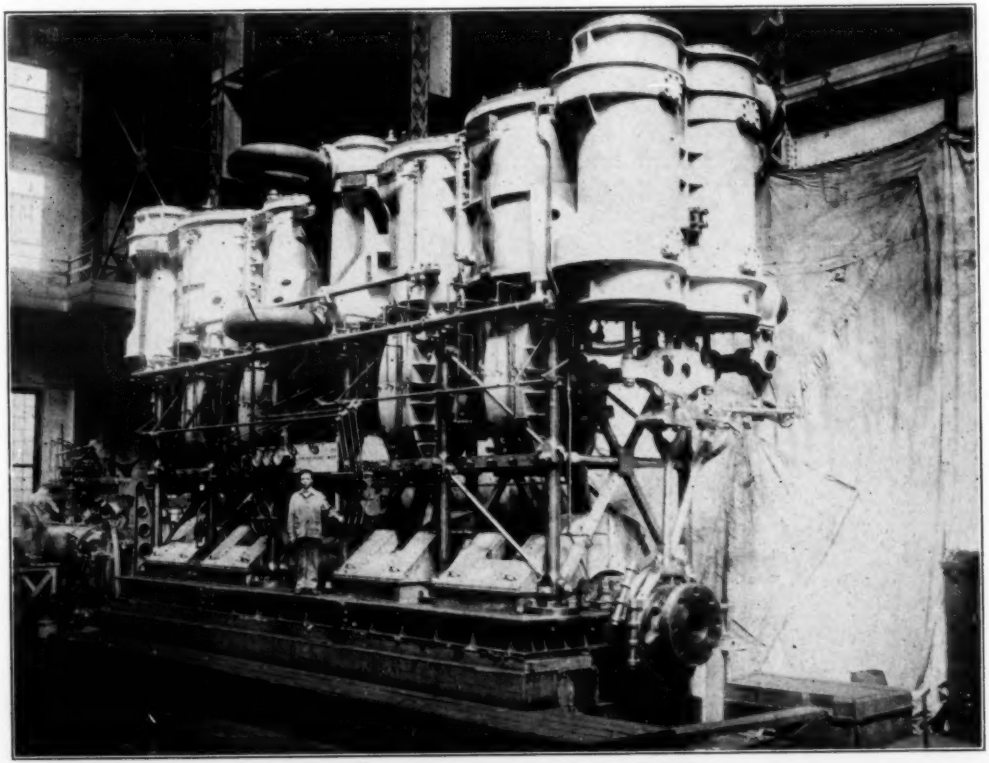
FORCED–LUBRICATION SYSTEM.

Each main engine is provided with its own system of forced lubrication, each system consisting of three steam oil pumps, one main-supply oil tank and one water and oil-settling tank, together with the necessary piping, valves and filters. The crank pits are of oiltight construction, fitted with an oil well, and are so designed that no bilge water or dirt can enter therein. An oil trough is thus formed at the base of each engine to catch all the oil. The tops of the crank pits are

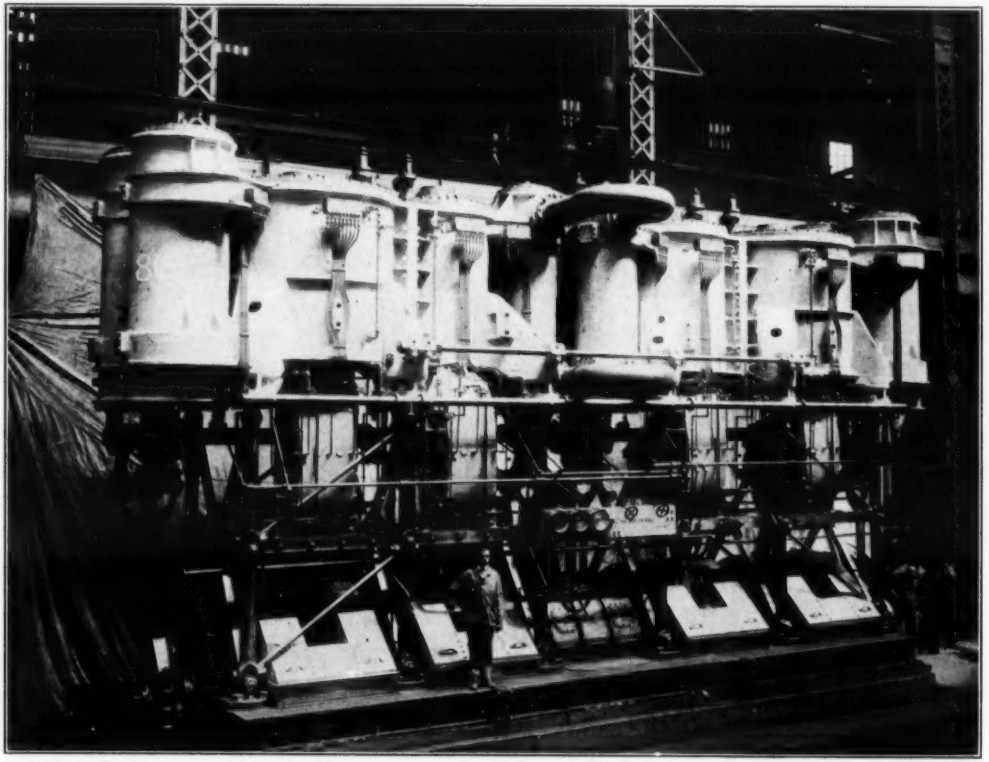
covered with a light sheet-iron casing to prevent splashing and waste of oil, but allowing sufficient clearance for the oscillation of the connecting and eccentric rods.

The pumps of each system are designated by A, B and C. Pump A draws oil from the main supply tank and discharges same through oil filters to the main bearings only. Pump B is arranged primarily to draw oil from the crank-pit oil well and discharge same through filters to the main supply tank. The filters are provided in triplicate, two intended to be used at a time, while the other is being overhauled and cleaned. This pump can also be used as an auxiliary on the water suction from oil well, with delivery to settling tank. Pump C is in the nature of a standby, principally intended to draw water and oil from the oil well and discharge same to settling tank, but it is also arranged to perform the duties of both the other pumps in case of emergency.

The plant operates as follows: Pump A draws the oil from the main supply tank and discharges same through a pipe leading to the main engine and through branch pipes to the main bearings. The branch pipes conduct the oil to the bearings through holes in the bearing caps. Annular grooves in the bearings provide for the proper distribution of the oil throughout the main bearings, the oil passing through a radial hole in each journal to the axial holes of the crank shaft. The openings at the end of axial holes are closed by oiltight cover plates. From the crank-shaft axial holes the oil is forced through radial holes to the crank-pin axial holes and through radial holes to the surface of the crank pins. The crosshead pins are provided with swivel joints and pipes and valves so arranged that either forced lubrication or the water surface can be used on the crosshead pins. All other parts of the engine are oiled in the usual manner, by gravity flow supplied from manifolds and cups. Ordinarily the crosshead pins are oiled in like manner, instead of being connected to the forced system. After performing its function the oil is forced out at the ends of the bearings and drains to the crank pit, which, as stated, serves the purpose of a trough for catching the oil. From the crank-pit oil well the oil is drawn by



STARBOARD ENGINE, U. S. S. "DELAWARE."



PORT ENGINE, U. S. S. "DELAWARE."

the pump B and delivered through oil filters to the main supply tank, thus completing its cycle, which is repeated indefinitely as described.

Whenever the oil shows signs of water, due to water-service waste and other sources, the pump C is put on the water suction from the oil well and discharges to the settling tank, from which, after allowing to settle, the oil is drained off to the crank pit and the water to the bilge. The settling tank has no suction connection to any pump.

Pressure gauges are provided to show the pressure at which the oil is supplied. Thermometers are also fitted in the main bearing caps for taking the temperature of the bearings, etc. Large storage tanks are provided for making up leakage and other waste and for replenishing the entire system when desired.

DATA OF MAIN ENGINES.

	H.P.		*I.P.*		*L.P.*	
Diameter of cylinders, each, ins.....	38½		57		76	
Stroke, each, inches.....	48		48		48	
Diameter of piston rods, each, ins...	8		8		8	
Balance pistons, each, inches.....	6½		6½		12	
VALVES AND VALVE SETTINGS.						
Number of piston valves, each.....	1		2		2	
Diameter of valves, each, inches.....	22		22½		32½	
Travel of valve, each, inches	9		10		11	
Side of valve on which steam is t'k'n	Inside.		Outside.		Outside.	
	Top.	*Bottom.*	*Top.*	*Bottom.*	*Top.*	*Bottom.*
Width of port, inches.....	$3\frac{7}{8}$	$3\frac{7}{8}$	$3\frac{1}{2}$	$3\frac{1}{2}$	$3\frac{5}{8}$	$3\frac{5}{8}$
Steam opening, linear, inches..	$3\frac{3}{32}$	$3\frac{7}{32}$	$3\frac{11}{32}$	$3\frac{5}{16}$	$2\frac{9}{16}$	$2\frac{9}{32}$
area, sq. inches.....	177	184	385	402	441	457
Exhaust opening, linear, inches.....	$3\frac{7}{8}$	$3\frac{7}{8}$	$3\frac{1}{2}$	$3\frac{1}{2}$	$3\frac{5}{8}$	$3\frac{5}{8}$
area, sq. inches..	221.8	221.8	425.2	425.2	624.4	624.4
Steam lap, inches.....	$1\frac{1}{16}$	$1\frac{9}{32}$	$1\frac{13}{64}$	$1\frac{1}{16}$	$2\frac{1}{16}$	$2\frac{7}{32}$
Exhaust lap, inches.....	$-\frac{5}{64}$	$+\frac{13}{64}$	$-\frac{1}{16}$	$+\frac{9}{32}$	$-\frac{1}{8}$	$+\frac{1}{8}$
Steam lead, linear, inches.....	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	I	$\frac{7}{8}$	$\frac{31}{32}$
Cut-off, per cent. of stroke, max.....	88.54	83.59	83.59	76.56	67.97	57.03
min.....	38.02	28.64	35.41	26.04	25.52	18.49
Diam. of crosshead bearings, ins....	12		12		12	
Length of crosshead bearings, ins...	11¼		11¼		11¼	
Diam. of crank-pin bearings, ins....	19		19		19	
Length of crank-pin bearings, ins...	22½		22½		22½	
Diam. of crank-shaft bearings, ins..	18½		18½		18½	
Length of crank-shaft bearings,						
Nos. 1, 2, 3, 4, 7, inches.....	...		...		23	
Nos. 5 and 6, inches.....	...		...		13½	
Axial hole, crank shaft, inches.....	...		...		11	
Length of connecting rods between centers, inches.....	...		...		96	
Ratio to crank.....	...		...		1:4	

CYLINDER CLEARANCES.

		In per cent. of volume.		*Linear, in inches.*	
		Top.	Bottom.	Top.	Bottom.
Starboard	H.P.	15.62	16.75	$\frac{5}{16}$	$\frac{9}{16}$
	I.P.	12.52	14.28	$\frac{5}{16}$	$\frac{7}{16}$
	F.L.P.	11.54	13.04	$\frac{3}{16}$	$\frac{7}{16}$
	A.L.P.	11.92	12.96	$\frac{7}{16}$	$\frac{19}{32}$
Port	H.P.	15.52	16.79	$\frac{7}{16}$	$\frac{9}{16}$
	I.P.	12.91	14.58	$\frac{7}{16}$	$\frac{7}{16}$
	F.L.P.	12.05	13.04	$\frac{7}{16}$	$\frac{9}{16}$
	A.L.P.	12.13	13.20	$\frac{19}{32}$	$\frac{11}{16}$

Cylinder constants. Mean for both ends, Km.

	H.P. Cylinder.	I. P. Cylinder.	F.L.P. Cylinder.	A.L.P. Cylinder.
Starboard	.138371	.306160	.546770	.546857
Port	.138063	.306041	.546770	.546770

I.H.P. of cylinder $= 2 \times Km \times Pm \times R$.

SHAFTING AND THRUST BEARING.

The thrust shaft is complete to the after section of the crank shaft and has thirteen collars, between which are fitted cast-steel horseshoes lined with white metal, which are hollowed for circulating water. The horseshoes may be adjusted by nuts on the side rods, which are secured to cast-iron pedestals at each end. These pedestals are bolted to the sole plate, and are capable of being moved in a fore-and-aft direction by adjusting wedges. The pedestals with their ends and side walls form a trough for an oil bath which has in it a cooling coil.

The line shaft is forged on the thrust shaft and continues to the stern tube, where it couples to the stern-tube shafting. The latter is supported by two stern-tube bearings, which are of brass and fitted with lignum vitae.

The inboard coupling of this shaft is made so as to permit the shaft to be drawn outboard.

The outboard body of the propeller shaft is covered with a watertight composition sleeve, shrunk on and then pinned.

Each propeller shaft is supported by strut and stern-bracket bearings fitted with lignum vitae bearing surfaces similar to those in the stern tube.

DATA FOR SHAFTING.

Diameter of thrust shaft, inches	17¼
Number of collars	13
Diameter of collars, inches	27
Width of collars, inches	2
Space between collars, inches	4
Length of thrust shaft, feet and inches	16–04½
Diameter of shaft beyond thrust, inches	17½
Length of stern-tube shaft, feet and inches	51–04
Diameter of stern-tube shaft, inches	18
Length of propeller shaft, feet and inches	51–05½
Diameter of propeller shaft, inches	18
Axial hole, inches	10½
Thrust shoes, effective surface, square inches	2,769
Length of thrust-shaft bearings (one), inches	18
line-shaft bearing (one), inches	26
stern-tube bearings, forward, inches	61
after, inches	61½
propeller-shaft bearings, forward, inches	35½
aft, inches	61½

PROPELLERS.

There are two three-bladed propellers, both outboard turning for ahead motion; the blades and hub are of manganese-bronze. The blades are a true screw at 19 feet 9 inches pitch, and the pitch is adjustable by having oval bolt holes in the flange of the blades. The hub is tapered to fit the shaft, and keyed, the nut being securely locked. All underwater fittings are covered by casings and fair-water sleeves filled with rosin and pitch to make them waterproof.

PROPELLER DATA.

Number of propellers	2
blades, each	3
Diameter, feet and inches	18–03
Pitch as set, feet and inches	19–09
adjustable from, feet and inches	18–09 to 20–09
Ratio of diameter to pitch	.924
Area, projected, square feet	85.4
helicoidal, square feet	105.84
disk, square feet	261.587
Height of lower tip of blade above keel, inches	6 3/16
Immersion of upper tip of blade at normal load draught	109½

BOILERS.

The boilers are of the Babcock & Wilcox type, with superheaters built and installed by that firm. The battery consists of fourteen boilers in four watertight compartments, two being installed in the after compartment and four in each of the three forward compartments.

ARRANGEMENT OF BOILERS.

The capacity of the boilers is such that all steam machinery can be run at full power with an average air pressure in the ash pits of not more than 2 inches of water. All pressure parts are made of open-hearth steel plate and seamless-steel tubes with no screw joints exposed to the fire. The general design of boilers is the usual type built by this well-known firm and are provided with superheaters.

Each boiler has the following fittings: One main steam stop valve closing toward the boiler and one dry pipe in the steam drum; one main-feed stop valve and check valve, with internal pipe; one auxiliary-feed stop valve and check valve, with internal pipe; one surface-blow valve, with internal pipe and scum pan, and two bottom-blow valves, with internal pipes; one main safety valve and one steam gauge; three gauge cocks to operate from the fireroom floor; two glass water gauges of the Klinger Reflex type, fitted with quick-closing devices; one salinometer pot, one drain cock and one air cock; one stop valve, three-quarters of an inch in diameter, for cleaning pipe connections; two cleaning hoes and lances; one connection for testing water; necessary pipes and fittings for attaching all the above to boilers. Zinc protectors with baskets for catching pieces of disintegrated zincs. There are air connections for blowing soot off the tubes.

The uptakes have a clear area at all points of one-seventh of the grate, and are provided with dampers and doors for examination of the interior. The space between the inside and middle sheets is open to the air space of the smoke pipe

at the top, and is provided with dampers at the bottom for the free circulation of air when forced draft is not used. These two sheets are free, one from the other, to allow for expansion. The space between the middle and outer sheets is filled with magnesia to prevent conduction of heat, and this lagging extends to the protective deck.

There are two smoke pipes, each about 91.5 feet in height above the grates. The forward pipe is 12 feet 4 inches, and the after pipe 10 feet 8 inches, inside diameter. The inner and outer casings extend to the top, and between these two casings is the usual air space. The weight of the smoke pipe is taken by the protective deck. The forward smoke pipe is connected with the uptakes of the eight forward boilers and the after smoke pipe with the uptakes of the six after boilers, as shown in sketch.

DATA FOR BOILERS.

Type	Babcock & Wilcox.
Number	14
Height, external, feet and inches	13–11¼
Length, external, feet and inches	11–06
Width, external, feet and inches	16–06
Number of furnaces, each boiler	1
furnace doors, each boiler	4
headers, each boiler	27
Total grate surface, square feet	1,439
boiler heating surface, square feet	55,749
superheater heating surface, square feet	6,194
heating surface, square feet	61,943
Ratio of G.S. to H.S. (exclusive of superheating surface), sq. ft.	1:38.6
Grates, length, feet	7
width, feet and inches	14-09
Test pressure, pounds	450
Working pressure, designed, pounds	295
Number of tubes, each boiler, 2-inch (8 B.W.G. thick)	795
4-inch (6 B.W.G. thick)	29
Smoke pipes, height above grate, feet	91.5
Area of smoke pipes, each, square feet, forward	119.46
after	89.36
Number of smoke pipes	2
Grate area, each group, square feet, forward	822.48
after	616.86
Kind of forced draft	Closed fireroom.

ELECTRICAL FIRING DEVICE.

There is installed in each fireroom an electrical indicator for regulating the intervals of firing the furnaces. The interval of firing the furnaces is controlled from a transmitter in the starboard engine room. A gong in each fireroom calls attention to the number displayed.

FUEL-OIL SYSTEM.

In addition to the usual coal-burning appliances, a complete oil-burning system has been installed, consisting of the necessary pumps, tanks, oil heaters and piping. Each boiler is fitted with three oil burners of the Schutte and Koerting type. The oil is stored in the double bottom of the engine room. In each engine room an oil pump takes the oil from the storage tanks and discharges into an 8,000-gallon settling tank, located aft and outboard of the port engine room. An electrical signaling device, operated from a float in the settling tank, is used in connection with regulating the oil pumps to keep the oil supply in the settling tank as near the normal level as possible. The signal consists of three electric lights, located near the pumps, which indicate the normal, low or high oil level. In each fireroom two small oil pumps are installed, which take the oil from an oil-suction main connecting with the bottom of the settling tank. These pumps discharge the oil through oil filters and oil heaters and then to the oil burners.

FUEL-OIL TANKS IN INNER BOTTOM.

Compartment.	Gallons.	Compartment.	Gallons.
C-92 . . .	13,952	C-96 . . .	13,372
C-93 . . .	13,952	C-97 . . .	13,372
C-94 . . .	8,184	C-98 . . .	15,140
C-95 . . .	8,184	C-99 . . .	15,140

MAIN STEAM LINE.

The main steam pipes are arranged symmetrically in two systems, one on each side of the ship. The valves and connections are such that the steam, after being generated in the

boilers, may be sent either through the superheater or, if desired, direct to the main steam pipe.

The branches from the boilers and the superheaters are 5½ inches diameter, and the main steam pipes, which are 8 inches diameter at the forward boilers, are increased from forward aft to 9½ inches, 10 inches, 11 inches, 12 inches and 13½ inches, respectively, the latter diameter continuing to the engine. Slip joints, bleeder pipes and separators are provided, as well as stop valves, which are suitably located in the lines of piping. There is provision for supplying live steam to the intermediate and the low-pressure receivers.

AUXILIARY STEAM LINE.

Forward of the connections from the forward boilers and the main steam pipe there is an auxiliary steam pipe, forming a connecting loop between the forward ends of main steam pipes.

In each engine room there is an auxiliary steam pipe connecting with the separator and passing along the outboard side of the engine rooms, and through the middle-line bulkhead, forming a connecting loop between the two sides of the ship.

The auxiliary steam pipe in the forward boiler room supplies steam to the machinery in the forward part of the vessel and the auxiliary steam pipes in the engine rooms supply steam to the machinery in the after part of the ship.

In connection with the auxiliary steam line there are the following systems:

Dynamo steam and exhaust piping;
Evaporator steam and exhaust piping;
Steam heating and pantry system;
Ice-machine steam and exhaust piping;
Anchor-engine steam and exhaust piping;
Steering engine;
Galley steam;
Laundry steam.

OTHER SYSTEMS OF PIPING.

Other systems of piping are :
Main-feed suction and discharge ;
Auxiliary-feed suction and discharge ;
Bilge suction and discharge ;
Reserve-feed tank connection ;
Distiller and evaporator piping ;
Fresh-water piping ;
Discharge from pump to fire mains ;
Sanitary system ;
Water service to main engines ;
Drains ;
Steam piping to sea valves ;
Refrigerator piping ;
Ice-machine water piping ;
Vapor piping ;
Safety-valve escape pipes ;
Indicator piping ;
Forced-lubrication piping ;
Fuel-oil piping.

TELEGRAPHS AND GONGS.

The engine-room telegraphs and gongs are of the usual type and are installed in two independent systems, one system being operated from the conning tower and the central station and the other from the bridges.

TELEPHONES AND SPEAKING TUBES.

Telephones of the loud-speaking type are installed, so that communication is obtained where needed. Speaking tubes communicate from the central station, from the bridges and from engine room to various other stations.

ASH HOISTS.

The ash-hoist engines are of the usual Williamson Bros. two-cylinder reversible type, and are located in the upper hatches. The ventilators contain the bucket guides, ropes, sheaves, etc. The hoists are operated from the main deck,

the bucket-guide rails extending from the bottom of the ventilator trunk to a point well above the main deck. There are trolleys at this deck for delivering ash buckets to the chutes at the ship's side. Each engine has a capacity of hoisting 300 pounds from the fireroom floor to the main deck in five seconds, with 150 pounds steam pressure.

The hoisting engines are of the reciprocating type, with a rope drum, fitted with a follow-up and reversing gear and an adjustable safety gear to prevent over winding and to stop the engine when the bucket reaches the fireroom floor.

DATA FOR ONE ENGINE.

Number of cylinders	2
Diameter of cylinders, inches	3½
Stroke, inches	4½
Diameter of drum, inches	5
Number of ash-hoist engines in the ship	4

ASH–EXPELLER SYSTEM.

In addition to the usual ash-hoisting engine and gear provided for hoisting ashes an ash-expeller system has been installed in each fireroom for expelling ashes through the bottom of the ship. Each system consists of an ash expeller, an expeller engine, one H.P. and one L.P. air compressor and two air tanks and necessary piping.

The method consists of an apparatus by which the ashes and clinkers are forced through the bottom of the ship by means of compressed air, and posseses none of the objections inherent to other methods. Figure 1 is a general arrangement of the ash expeller, as it has been named.

The expeller proper consists of a hopper to receive the ashes and clinkers opening into a crusher, which breaks up the large clinkers. Below the crusher is a drum revolving horizontally in a watertight casing or barrel very similar to a single-ported plug in a large taper cock. This drum makes about eighteen revolutions per minute. As it revolves the inside of the drum is alternately in communication with the chamber below the crusher and the discharge opening

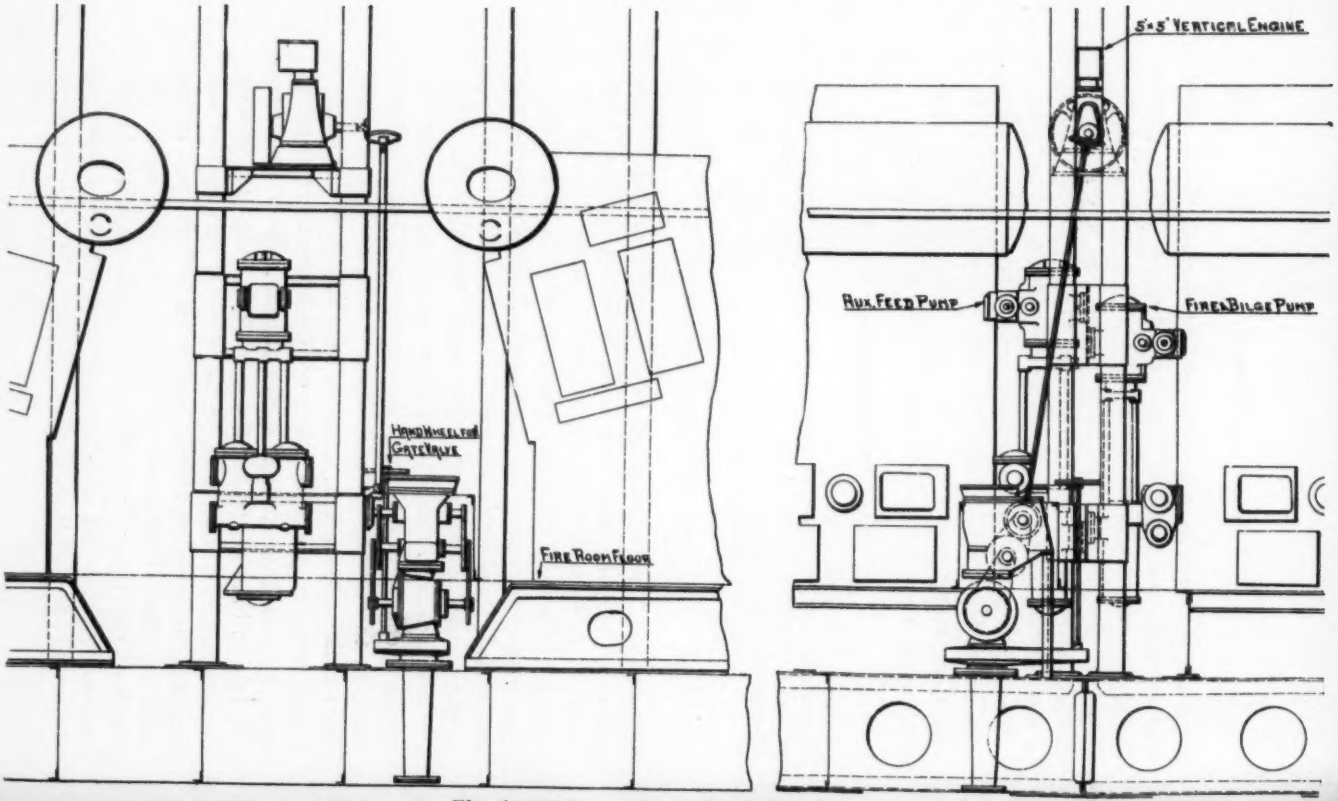


Fig. 1.—Arrangement of Ash Expeller.

through the bottom of the ship. A sepecially-constructed gate valve is fitted immediately below the expeller between same and the cast-steel discharge pipe going through the vessel's bottom. This valve is closed when the expeller is not in use; the hopper is also provided with a watertight door. About from fifty to sixty cubic feet of free air compressed to seventy pounds, are required for each expeller. Air is supplied by a specially-constructed steam-driven vertical compressor.

The action of the machine is as follows: The ashes and clinkers on passing through the crusher fall into the chamber immediately over the revolving drum; as the opening in the drum comes round opposite the chamber the ashes and clinker fill the interior of the drum by gravity. As the drum revolves the opening in same moves round to the solid side of the expeller casing or barrel, but before it reaches the discharge opening at the bottom, compressed air is turned on to the interior of the drum, with the result that when the drum and discharge opening come together, the pressure of the air in the drum, being much greater than the water pressure, the air suddenly expands, forcing out the water from the discharge pipe and carrying the ashes with it with such force that it is swept completely clear of the bottom of the vessel. Then as the drum continues to revolve, the opening in same is again covered by the solid side of the casing, the compressed air is shut off, and the air in the drum allowed to escape, so that when the opening again comes round to the chamber below the crusher the drum is ready to receive a fresh charge.

Fig. 1.—Arrangement of Ash Expeller.

FORCED-DRAFT BLOWERS.

Four forced-draft blowers are installed for each of the three forward firerooms and two for the after fireroom. They are located on the protective deck in separate airtight compartments above the center of the firerooms.

The fans are of the Sirocco type, and each blower is driven

by a General Electric motor. The air is supplied from the fireroom ventilators, which are closed at the bottom when under forced draft.

DATA FOR ONE BLOWER FAN.

Diameter of fans, inches	27
Number of blades	64
Area of induction nozzle, square inches	452
eduction nozzle, square inches	864

FEED AND FILTER TANK.

A feed and filter tank of 5,250 gallons capacity is located on the outboard side of the after end of each engine room. It is rectangular in shape and the top is arranged as a filter, with a capacity of 900 gallons; the filtering material is "Loofa," which is placed between the perforated plates. This filtering chamber is divided by vertical division plates so arranged that the water flows under and over in succession. All the material is of steel which has beed galvanized before assembling.

TANKS.

There are oil tanks of 2,000 gallons capacity, there are tallow tanks of 300 pounds capacity, and two waste tanks of 500 pounds capacity each.

MACHINE SHOP.

The maching shop is located amidship, on a level with the battle gratings and in a compartment of its own between the engine-room hatches. Each machine is directly driven by its own motor.

The following machines are installed:

One 25 × 39-inch extension gap lathe (12 feet 6 inches).
One 14-inch engine lathe (7 feet bed).
One 14-inch bench lathe (5 feet bed).
One 16-inch shaper.
One 28-inch vertical drill press.
One 16-inch vertical drill press.

One universal milling machine.
One 12-inch emery grinder.
One portable cylinder-boring machine.
Six bench vises.

All tools are complete and provided with the most modern attachments, including scroll and drill chucks, index heads, automatic crossfeed, swivel tables, pipe vises, etc., and all necessary tools, drills and cutters.

HEATING SYSTEM.

The heating system is divided into sections or groups, each group having a supply manifold and a drain manifold. The supply manifold of each group takes steam from the auxiliary steam pipe, with a stop valve, an adjustable reducing valve and a steam gauge at the connection with the auxiliary steam pipe, the reducing valve being set at 50 pounds. There are eleven separate and independent circuits supplying steam to the radiators in different parts of the ship. The radiator manifolds are each drained by an independent trap.

All traps discharge through main steam-drain line into the feed tank, and are fitted with suitable by-pass and stop valves to discharge into the auxiliary condenser.

The heater-circuit steam and drain pipes are of seamless-drawn brass (iron-pipe size), suitably connected by composition fittings, made tight when passing through bulkheads by stuffing boxes, and located as near to the door as possible when passing through storerooms.

Provision for ample expansion has been made by fitting horizontal copper U bends having the same internal diameter as the connecting pipes. The radiators consist of brass pipes connected by return bends with reduced ends. They are fitted generally with drain, supply stop valves, triangular stems for socket keys, protected by stem guards, but have hand wheels in officers' quarters.

Wherever areas over ten square feet are called for the radiator coils are divided into two parts, unions being fitted on

outlet ends of valve to permit of unrestricted removal of coils. In order to expedite service and repairs all heaters and radiators are tagged by suitable brass tags bearing numbers, odd and even, from forward to aft, on each deck, according to location on starboard and port side, respectively.

REFRIGERATING PLANT FOR COLD STORAGE.

There are two Allen dense-air ice machines, each capable of producing the cooling effect of two tons of ice per day. The cooling pipes from the machines are led into the ice tank, the scuttle butts and the cold-storage rooms.

Valves are provided, in accordance with the Bureau of Steam Engineering Standard Arrangement of Valves, in the cold-air pipes of refrigerating plant, so that the air may go to the cold-storage room direct or through the ice-making tank, and thence to the cold-storage room and scuttle butts, and also from the ice-making tank direct to the scuttle butts.

MAGAZINE–COOLING SYSTEM.

There are three Allen dense-air ice machines, each capable of producing the cooling effect of three tons of ice per day, for cooling the magazines. One of the machines is located forward on the protective deck and cools the air to the forward magazines; two machines are installed in the after part of the ship, on the protective deck, and cool the air to the after magazines.

Suitable air boxes or tanks are installed for the purpose of cooling the air before it passes to the magazines. Each air box is fitted with a cooling coil through which the cold air from the ice machines circulates. Suitable supply and return air ducts are provided, leading the air from the air boxes to the magazines and back to the air boxes, the air being used over and over again. Blowers are used for delivering the air from the air boxes to the magazines.

AIR-COMPRESSOR PLANT FOR PNEUMATIC TOOLS.

There are three Westinghouse air compressors in the after part of each engine room for use in running pneumatic tools in the steam engineering department, for blowing soot off the boiler tubes, and for the gas-ejecting system for the guns.

Each compressor has a capacity of about 50 cubic feet of free air per minute at 125 pounds pressure. The compressors are driven by steam.

CONDENSERS.

Main Condenser.—There is one main condenser in each engine room, 7 feet 9 inches inside diameter. The shell is $\frac{7}{16}$ inch thick, with three double-butt joints, and with circumferential T-bar stiffeners.

The water chests are cast $\frac{3}{8}$ inch thick. The forward chest, being the one for the entrance and exit of circulating water, has a horizontal division plate in the middle fitted with valves, which, when open, allow the circulating water to pass overboard direct, the valves being worked by a lever on the outside of the condenser, and it has nozzles for inlet and outlet of circulating water 21 inches in diameter.

The condenser-tube sheets are rolled, each in one piece. They are 1 inch thick, with smoothly finished holes for the tubes, tapped and fitted with screw glands for packing the tubes.

There are 5,546 seamless-drawn tubes in each condenser, $\frac{5}{8}$ inch outside diameter, No. 16 B.W.G. in thickness. The tubes are 13 feet long between tube sheets and are spaced $1\frac{5}{16}$ inch between centers. They are supported in the center by ferrules, $\frac{3}{4}$ inch long, inserted in a supporting plate. Baffle plates are fitted to direct the steam over all the tubes.

The cooling surface for each condenser is 11,788 square feet, measured on the outside of the tubes.

The condensers are supported by angle plates riveted between circumferential angle-bar stiffeners.

There are riveted to the shell of each condenser two cast composition nozzles with faced flanges, 30 inches diameter of

opening, for attachment of the main exhaust pipes. The after nozzle is provided with a faced flange, 9 inches in diameter of opening, for the auxiliary exhaust pipe.

Cast-steel flanges are riveted to the shell of each condenser and properly faced for an air-pump suction pipe 12 inches in diameter and for a manhole at the bottom 12 inches in diameter. There is a 1-inch connection in the bottom manhole for cleaning the tubes by boiling. Drain cocks are provided with pipes leading to the bilge.

A copper tank is provided for admitting an alkaline solution into each condenser, the tank having a capacity of five gallons. Zinc protectors and safety valves are fitted.

The material of the condenser is as follows: Shells, steel, class "B" boiler plate. Baffle plates, steel, class "C" boiler plate. Tubes, composition: copper 70, tin 1, zinc 29 per cent. Tube sheets and supporting plates are as near as possible the same material as the tubes. Glands are of tubing, and of the same composition as the tubes. Water chests are of composition. All bolts are of bronze.

MAIN AIR PUMPS.

There is one independent air pump for each main engine, which is capable of maintaining a vacuum of 25 inches at full power. Each is double, vertical and single acting, with two inverted steam cylinders. The suction opening is 12 inches and discharge is 11 inches in diameter. The water cylinders are of composition and the steam cylinders of cast iron.

MAIN CIRCULATING PUMPS.

The circulating water for the main condenser is supplied by a centrifugal pump having double inlet valves, and it is driven by an independent compound engine of sufficient power to discharge 15,000 gallons of water per minute when making 265 revolutions per minute.

AUXILIARY AND DYNAMO CONDENSERS.

The material of these condensers is as follows: Shells, steel, class "B;" tubes, composition: copper 70, tin 1, zinc 29 per cent.; tube sheets, as near as possible of the same material as the tubes; heads, composition.

The cooling surface of each auxiliary condenser is 600 square feet and of each dynamo condenser 1,206 square feet, measured on the outside of the tubes. The tube sheets are one inch thick. The diameter and spacing of the tubes and the packing are the same as in the main condensers.

In each engine room there is an auxiliary condenser connected from the auxiliary-exhaust pipe to all the auxiliary machinery. Each condenser has a combined air and circulating pump.

For each dynamo room there is an auxiliary condenser for the exclusive use of the dynamo engines. The circulating water is supplied by a centrifugal pump, driven by an independent single-cylinder engine. Each condenser has its own independent air pump and feed pump. Each air pump discharges into a small feed tank, and the feed pumps discharge into a pipe leading to the main feed tanks in the engine rooms.

EVAPORATING AND DISTILLING PLANT.

The evaporator and distilling plant is placed on the protective deck. There are four evaporators and four distillers, with their accessories.

The evaporators have a combined capacity of 25,000 gallons of water per twenty-four hours.

The distillers have a combined capacity of 25,000 gallons of potable water per twenty-four hours.

The evaporators take steam from the auxiliary steam pipe, and the coil-drain pipes lead through a by-pass, automatic traps, to the feed tanks. The evaporator-feed and fresh-water pumps take steam from the evaporator coils, as well as from the auxiliary-steam pipe.

The shells of the evaporators have connection with valves

and pipes for directing the steam into the distillers and into the auxiliary-exhaust pipe.

The feed water for the evaporators is taken from the circulating pipe, after it has passed from the sea and through the distillers.

There are blow pipes of ample size, so arranged that the evaporators may be blown out when working under pressure.

The distiller circulating pumps discharge to the distillers, the water passing overboard or into the sanitary system at will. There is also a direct connection from this pump to the flushing pipes, and the relief valve for the distiller is so placed as to act for both the distillers and the flushing system.

In addition to the pump connection for distiller circulating provision is made for circulating water through the distillers, from a conveniently located discharge pipe from a pump discharging to the fire main.

A small reservoir tank, fitted with a removable cover, is placed in the fresh-water pump suction between the pump and the distillers.

A water meter, without lead, is placed in the pipe connecting the reservoir tank with the bottom of the distillers.

Evaporators.—There are four horizontal evaporators, each having 312 square feet of tube-heating surface. The tubes are of brass, 1½-inch gas-pipe size, two tubes being connected by a return bend.

The tubes are secured to the tube sheets so that adequate provision is made for expansion, and the tubes are so arranged that their removal will leave the shell accessible in all its parts for scaling. All tubes are easily removable for cleaning and repairs.

Distillers.—There are four distillers, each having 125 square feet of tube-cooling surface, measured on the outside of the tubes.

The tubes are straight, ⅝ inch outside diameter, tinned on both sides and well expanded and sweated into tube sheets. Provision is made for the expansion of the tubes by the use of flange tube sheets working in a stuffing box.

FEED-WATER HEATER.

In each engine room there is a feed-water heater with all necessary fittings, complete.

The heating surface of each heater is 1,050 square feet, measured on the outside of the tubes. They are of the return-flow type, located on the discharge side of the main feed pumps.

The heating agent is the auxiliary exhaust steam. The tubes are ⅝-inch outside diameter, No. 16 B.W.G.

PUMPS AND PUMP CONNECTIONS.

The pumps and pump connections are in accordance with the tables I and II on the following pages (1250 and 1251).

ELECTRIC PLANT.

There is one dynamo room, forward of the forward boiler room and below the protective deck.

The generator installation consists of four eight-pole, compound-wound, 300-kilowatt, G. E. generators, driven by Curtis turbines. Each of these generators is capable of delivering 2,400 ampères of current at 125 volts when running under full load at 1,500 revolutions per minute, with a shunt current of 25 ampères.

The dynamo steam lines connect with the auxiliary steam line in the forward boiler room. The piping is so arranged that steam may be taken from either side of the ship to either starboard or port dynamos. The steam pressure used is 200 pounds, which is regulated by means of a Foster reducing valve.

In each steam line to the dynamo room there is a separator and a trap for separating water from the steam and draining it off. The generator and forward light and power distribution boards are located in compartment A–61 port; the after distribution board is located in compartment C–35.

The generator panels contain circuit breakers, volt meter,

Table I.—Pumps.

Auxiliary pumps.	Number.	Type, make and location.	Steam cylinders. Number.	Diameter, inches.	Dia. piston rod, ins.	Stroke, inches.	Const. km. (mean of both ends).	Water cylinders. Number.	Diameter, inches.	Dia. piston rod, ins.	Stroke, ins.
Main air	2	Blake, twin, vertical bucket, single-acting. One in each engine room.	2	14	$2\frac{1}{4}$	21	.00806	2	35	$2\frac{3}{4}$	21
Main circulating	2	Centrifugal, driven by compound engine. One in each engine room.	2	H.P. 11 L.P. 22	$2\frac{1}{4}$ $2\frac{1}{4}$	10 10	.00235 .00955	.. ..	... ...	... ...	.. ..
Main feed	4	Blake, vertical piston, double-acting, single. Two in each engine room.	1	$13\frac{1}{2}$	$2\frac{1}{2}$	18	.0064	1	$9\frac{1}{2}$	$2\frac{1}{2}$	18
Auxiliary feed	4	Blake, vertical piston, double-acting. One in each fireroom.	1	$13\frac{1}{2}$ 10	$2\frac{1}{2}$ $1\frac{7}{16}$	18 12	.0064 .00235	1	$9\frac{1}{2}$ 7	$2\frac{1}{2}$ $1\frac{7}{16}$	18 12
Magazine insulation	2	Blake, vertical piston, double-acting, single. One in each engine room.	1	6	$1\frac{7}{16}$	8	.00055	1	8	$1\frac{7}{16}$	8
Fire and bilge	8	Blake, vertical piston, double-acting, single. One in each fireroom, two in each engine room.	1	12	$2\frac{1}{4}$	18	.00513	1	10	$2\frac{1}{4}$	18
Engine-room oil	2	Blake, horizontal piston, with hand attachment. One in each engine room.	1	2	$\frac{1}{2}$	$2\frac{3}{4}$	...	1	$1\frac{1}{2}$	$\frac{1}{2}$	$2\frac{3}{4}$
Auxiliary condenser	2	Blake, combined horizontal piston, single. One in each engine room.	1	6	$1\frac{1}{2}$	12	.00086	1	10	$1\frac{1}{2}$	12
Dynamo cond. circ	2	Centrifugal, driven by single-cylinder engine. One in each dynamo room.	1	5	$1\frac{1}{4}$	5	.00024	..	...	...	..
Evaporator feed	1	Blake, vertical piston, single. Evaporator room	1	$4\frac{1}{2}$	$\frac{13}{16}$	6	.00023	1	5	$\frac{13}{16}$	6
Distiller fresh-water	1	Blake, vertical piston, single. Evaporator room	1	$4\frac{1}{2}$	$\frac{13}{16}$	6	.00023	1	5	$\frac{13}{16}$	6
Distiller circulating	2	Blake, horizontal piston, double-acting. Evaporator room.	1	12	$1\frac{3}{4}$	12	.00339	1	14	$1\frac{3}{4}$	12
Dynamo cond'r air	2	Blake, twin, vertical bucket, single-acting. One in each dynamo room.	1	8	$1\frac{1}{2}$	12	.0015	1	16	$1\frac{1}{2}$	12
Dynamo filter tank	2	Blake, vertical piston, double-acting, single. Dynamo room.	1	$4\frac{1}{2}$	$\frac{13}{16}$	6	.00023	1	5	$\frac{13}{16}$	6
Engine-room fuel-oil	2	Blake, vertical piston, double-acting, single. One in each engine room.	1	7	$1\frac{7}{16}$	12	.00114	1	7	$1\frac{7}{16}$	12
Fireroom fuel-oil	8	Blake, vertical piston, double-acting, single. Two in each fireroom.	1	$4\frac{1}{2}$	$\frac{13}{16}$	4	.00016	1	$2\frac{3}{4}$	$\frac{13}{16}$	4
Forced lubrication	6	Blake, vertical piston, double-acting, single. (4 ps. Three in each engine room (2 ps.	1 1	6 $7\frac{1}{2}$	$1\frac{7}{16}$ $1\frac{7}{16}$	8 8	.00055 .0009	1	7 $8\frac{1}{2}$	$1\frac{7}{16}$ $1\frac{7}{16}$	8 8
Oil-settling tank drain	1	Blake, vertical piston, double-acting, single. One in port engine room.	1	$4\frac{1}{2}$	$\frac{13}{16}$	4	.00016	1	$2\frac{3}{4}$	$\frac{13}{16}$	4

Table II.—Pumps and Pump Connections.

		Three in each engine room	1	2 ps., Blake, vertical piston, double-acting, single.	1	7½	1 7/8	8	.0009	1	8½	1 7/8	8
Oil-settling tank drain.	1	in port engine room.	1	One	1	4½	1 1/2	4	.00016	1	2¾	1 1/2	4

Table II.—Pumps and Pump Connections.

No.	Pumps.	Size.	Kind.	Suction pipes from—	Ins.		Discharge pipes to—	Ins.	Location.
2	Main air	Inches. 2 14, 2 35 x 21.	Twin, vert. bucket, single-acting.	12	Condenser	10	Main feed tanks		1 in each engine room.
2	Main circulating	11x22x10	Vertical compound, centrifugal.	15	Sea	20	Condenser		1 in each engine room.
				15½	Main drain	20	Overboard.		
4	Main feed	13¼x9½x18	Vert. piston, double-acting, single.	5	Feed-tank con'g pipe	4½	Main feed-disch. main		2 in each engine room.
				5	Channel ways	4½	Nozzle.		
3	Auxiliary feed	13½x9½x18	Vert. piston, double-acting, single.	3	Bottom blows	4½	Auxiliary feed main		1 in each boiler room, Nos. 1, 2 & 3.
				5	Feed suction pipes	3	Overboard.		
				4	C. and R. pipes from ship's side.*	3½	Reserve feed tanks.*		
					Reserve feed tanks. Thro' feed suct. pipe	4½	Nozzle.		
1	Auxiliary feed	10x7x12	Vert. piston, double-acting, single.	2	Bottom blows	3	Auxiliary feed main		1 in boiler room No. 4.
				3½	Feed suction pipe. Reserve feed tanks. Thro' feed suct. pipe.	2	Overboard.		
4	Fireroom fire and bilge.	12x10x18	Vert. piston, double-acting, single.	5	Fireroom floors	5	Fire main		1 in each boiler room.
				5	Sea	5	Overboard.		
				5	C. & R. dra'ge mans	2½	Hose connection.		
				1½	Hose connection	5	Sanitary.†		
4	Engine-room, fire and bilge.	12x10x18	Vert. piston, double-acting, single.	5	Sea	5	Fire main		2 in each engine room.
				5	Engine-room bilge	5	Overboard.		
				5	C. & R. dra'ge mans	2½	Hose connection.		
				1½	Hose connections	5	Sanitary.		
2	Shaft bilge	8x8	Plunger, single-acting.	4½	Engine-room bilges	4½	Overboard		1 in each engine room.
				4	Water serv. discharge.				
2	Distiller circulating.	12x14x12	Hor. piston, double-acting, single; or turbine.	5	Sea	5	Distillers		Evaporator room.
						5	Fire main.		
						5	Flushing main.		
2	Auxiliary air and circulating.	6x10x10x12	Comb'd hor. double-acting, single.	5	Auxiliary condenser	3½	Main feed tanks		1 in each engine room.
				5½	Sea	5½	Overboard.		
2	Dynamo air	8x16x12	Blake featherw'ght.	7	Dynamo condenser	6	Dynamo feed tanks		In dynamo room.
2	Dynamo circ'l'g	5x5	Vert. centrifugal	7	Sea	7	Overboard.		
1	Evaporator feed	4½x5x6	Light service, vert., dbl.-act'g, single.	2	Sea	2	Evaporators		Near evaporators.
				2	Dist'r circ. water disc.				
1	Distiller fresh-water.	4½x5x6	Vert. piston, light service.	2	Bottom of distillers	2	Fresh-water tanks		Evaporator room.
						2	Main feed tanks.		
						2	Res. feed-water tanks.		
2	Engine-room oil	2x1¼x2¾	Piston, with hand attachment.	1	Oil-storage tanks	¾	Distrib. tank, or 20-gal. tanks‡		1 in each engine room.

* In No. 3 boiler room only. † In Nos. 1 and 2 boiler rooms only. ‡ With overflow pipe from tank to pump suction.

ammeter and lighting and power, common negative and equalizer switch for each machine.

On the right of generator panel is the forward feeder panel, which supplies light and power through circuit breakers to the forward distribution board. Ammeters on this panel show total lighting and total power loads on forward distribution boards.

On the left of the generator board is the after feeder panel, which supplies the after distribution board through circuit breakers.

The power and turret panel of forward distribution board is located just forward of forward feeder panel, and supplies various power feeders through double-pole single-throw switches; turrets Nos. 1 and 2 are supplied by circuit breakers on this board.

The lighting panels are located forward and at right angles to the power panel. These panels also supply the searchlights and contain the necessary appliances for the control of same.

The after distribution board contains lighting, power and turret panels. The lighting panel is supplied from the after feeder panel on generator board by duplicate bus feeders connected to the two double-pole single-throw switches.

At the bottom of the lighting panel one feeder runs on the port side of the ship and the other on the starboard side. Either one, or both together, may be used to supply light to after board as desired.

Power and turret supply is obtained in a similar manner, the large switches at the bottom of power panels being for that purpose.

The various section switches connect to the light and power feeders supplied by the after board. The double-pole circuit breakers control supply of current to turrets Nos. 3, 4 and 5.

The lighting panel has an ammeter showing total lighting on after board. On the power panels are three meters—an ammeter showing total power load, a ground-detector volt meter with four-point switch, and a 150-volt meter—and by

means of its four-point switch shows the voltage of the power and lighting panels respectively.

On the middle panel is a single-point switch connected to ground-detector lamps. There is also a lamp ground detector on the lighting panel.

The forward distribution board is similarly supplied with ground detectors.

ANCHOR WINDLASS.

The anchor windlass is of the vertical type, and has two vertical shafts driven by worm gearing direct from a worm located on the crank shaft of the engine, without the intervention of counter shafts or beveled gears. Each shaft carries on its upper end, above the forecastle deck, a wildcat or locking gear, complete.

The arrangement is such that the wildcats can be operated together or independently of each other. The wildcats revolve in a horizontal plane, taking in each bower chain on the outboard side and the sheet chain on the inboard side of the port wildcats only. The vertical shafts have couplings, and the lower end is keyed and supported by steady bearings.

The wildcat is cast from the best open-hearth steel to suit the 2¾-inch cable, as manufactured by the Government.

Each wildcat is fitted with a positive locking device worked by raised cams on the periphery of the locking rim, and slotted keys operated by means of a lever. The entire operation of locking or unlocking the wildcat can be accomplished by one motion of the lever through an angle not exceeding sixty degrees.

A friction-band brake is fitted to each wildcat. The brakes have sufficient surface and ample strength in all parts to ride by with the windlass unlocked. Each end of the friction band is provided with a compressor, so that the bower chain can be checked when running out. An additional compressor is fitted to the friction band of the port wildcat to check the spare chain.

The compressor is controlled by a hand wheel set close to the wildcat.

The windlass engine is designed for a working steam pressure of 150 pounds per square inch, but is able to withstand the full boiler pressure.

STEERING GEAR.

The main steering arrangement consists of a right-and-left-handed forged-steel screw, connected through bushed cast-steel driving nuts and forged-steel side links direct to the crosshead on the rudder stock. The right-and-left-handed screw proper is connected by a lug clutch to another section of shafting, which shafting connects to the steering-engine crank shaft through a pair of herring-bone gears, the large gear being provided with a sliding clutch for throwing the engine in and out of gear. By throwing out this sliding clutch the steering engine may be thrown out of gear, in which condition the hand-steering wheels in the steering-engine room may be used for steering through the screw by throwing in clutch at forward end of the screw shaft; or, by leaving both of these clutches out and throwing in clutch on hand-steering wheel shaft, the rudder may be operated through the relieving tackle direct from hand-steering wheels to crosshead.

To put the rudder from hard over to hard over through the working angle of 70 degrees requires about 18 turns of the regular hand-steering wheels forward, when using the wire-rope transmission, 8 turns when using telemotor, and $10\frac{3}{4}$ turns of the trick wheel at the steering engine; $49\frac{1}{2}$ turns of the hand-steering wheels in the steering-engine room, when working through the relieving tackle; $17\frac{1}{4}$ revolutions of the screw shaft and $61\frac{1}{4}$ revolutions of the engine.

The rudder, which is of the balanced type, has an area of about 337 square feet. The main frame is a steel casting, with wrought-steel boundary frame and cross bracing of flanged plates and angles, the interior being filled with white pine

and the interstices with a mixture of half pitch and half tar, the whole being covered with 15-pound plating.

The weight of the rudder is taken on a bearing at the top of the stern casting, with a floating friction ring between the crosshead casting and stern casting. The crosshead casting has a horizontal, circular bearing surface at its upper end for the friction-band brake, which is operated by a hand wheel on the forward side of bulkhead No. 118½ through a long connecting rod and a bell-crank lever.

The steering engine is 17×14 inches, double-cylinder, combined hand and steam-steering engine, manufactured by the Hyde Windlass Co., of Bath, Maine. It is designed for a working pressure of 150 pounds per square inch, but is capable of withstanding full boiler pressure. The steam pressure for this engine is regulated by means of a reducing valve located on the after side of bulkhead No. 84 in the port engine room.

The operation of the engine by steam may be controlled either through wire-rope transmission from the steering stations forward, or by the hand wheel on the forward end of the engine. The engine is also operated by a telemotor located in the central station.

Steering Stations, Forward.—The steering stations forward are located in the communication room, conning tower and upper conning bridge. By means of a double-jaw-type clutch the communication-room wheel may be thrown into gear and the other two thrown out, or *vice versa.* In using the wheel in the upper conning bridge the conning-tower wheel may be thrown out by moving it aft on its shaft. In using the conning-tower wheel the upper conning bridge may be thrown out by disconnecting the shafting just above the conning-tower steering standard.

All shafting, except within twelve feet of compass, is cold-rolled steel, and all gears are composition with cut teeth.

Steering wheels are arranged so that when the tops of wheels go to port the helm goes to starboard and bow goes to port.

Electric helm indicators and electric steering-telegraph

transmitters are located in the communication room and upper conning bridge.

Steering Stations, Aft.—The after stations are both in the steering-engine room. One consists of the trick wheel connected directly to the operating shaft of the engine, the other consists of four large hand wheels on the same shaft, properly fitted, as described, with large clutches for steering, either through the screw shaft direct or through the relieving tackle.

The relieving tackle consists principally of rope drum, operating by the hand-steering wheels through a pair of bevel gears and a worm and worm wheel, on which is wound 1-inch diameter wire rope, one end of rope leading out to port and one end to starboard, and thence both over sheaves and through stuffing boxes in bulkhead No. 118½ and operating on the side rods through one single and one double sheave on each side of ship, the standing part of all being connected by turnbuckle to special stiffeners on after side of bulkhead No. 118½.

Both the trick wheel on the engine and the large steering wheels are so arranged that when the top of the wheel goes to port the bow goes to port also.

An electrical helm indicator and an electrical steering-telegraph indicator are located in the steering-engine room.

DESCRIPTION OF VENTILATION.

On this vessel all the compartments below the gun deck and the captain's cabin, the ward-room dining room, the armory, the bakery, the various crew spaces, water closets and wash rooms on the gun deck are provided with artificial ventilation.

Ventilation System.—Either supply—exhaust or both combined, but suitable to the requirements in each particular locality—are provided for the compartments in the various subdivisions of the ship, thus providing a positive circulation of air by means of electric blowers.

All estimates for changing air in the different compartments are based on the gross volume of the compartments.

In the following table the capacities of the blowers are given in cubic feet per minute at one-ounce pressure. All blowers are of the converted steel-plate type.

The air is renewed in the various spaces approximately as follows:

Officers' quarters and crew space, berth deck outside of armor bulkheads, in from ten to twelve minutes.

Officers' quarters and crew space, berth deck inside of armor bulkheads, in about four to six minutes.

Water closets and crews' head in about four minutes.

Wash rooms one to three minutes.

Sick-bay quarters in about eight minutes.

Store rooms and passages in eight to twelve minutes.

Magazines in from four to eight minutes.

Engine rooms and steering compartments in about two to four minutes.

Evaporator rooms in about two minutes.

Switch and distribution-board room and central station in about six minutes.

Dynamo rooms in about three-fourths of a minute.

Turrets in about two minutes.

This ventilation is provided by (24) electrically-driven fans, manufactured by the General Electric Co. Size, type and location are given in table.

None of the principal longitudinal or transverse water-tight bulkheads of the ship have been pierced by the ventilation ducts where possible to avoid it.

The 12-inch and 5-inch powder magazines are fitted with special ventilation, operated on both the closed and open systems. When the system is closed the air is drawn over a refrigeration coil and supplied to the magazines, the exhaust ducts from the magazines discharging back into the coil box. When operating on the open system the coil box is cut out and the air supplied from the open and exhausted through a cowl on the weather deck. Under the closed system a temperature of 70 degrees is to be maintained in the magazines.

Where ducts pass through protective deck they are made

VENTILATION SYSTEMS.

Fan No.	Spaces ventilated.	Blower.		Motor.		Location.			Type.
		Bur. No.	Capacity.	H.P.	R.P.M.	Deck.	Side.	Frame.	
1	Crew's W. C., paints and oils	...	2,500	1½	880–1,140	Main.	S.	5–6	Exh.
2	Quarters and stores forward 23	...	2,500	1½	880–1,140	Main.	P.	16–17	Sup.
3	12-inch turret, No. 2	...	2,500	1½	880–1,140	Main.	C. L.	U. T.	Sup.
4	12-inch turret, No. 1	...	2,500	1½	880–1,140	Gun.	C. L.	U. T.	Sup.
5	Sick bay	...	2,500	1½	880–1,140	Gun.	S.	42–43	Exh.
6	Dynamo room	...	10,000	6½	440–575	Gun.	P.	42	Sup.
7	Dynamo room	...	10,000	6½	440–575	Gun.	S.	42	Sup.
8	12-inch turret, No. 3	...	2,500	1½	880–1,140	Gun.	C. L.	U. T.	Sup.
9	Magazine-cooling system	...	2,500	1½	880–1,140	Gun.	S.	83	Closed.
10	Officers' W. C. and laundry	...	2,500	1½	880–1,140	Gun.	P.	93–94	Exh.
11	12-inch turret, No. 4	...	2,500	1½	880–1,140	Gun.	C. L.	U. T.	Sup.
12	12-inch turret, No. 5	...	2,500	1½	880–1,140	Gun.	C. L.	U. T.	Sup.
13	Magazines and stores, 23–33	...	5,000	3	615-810	Berth.	P.	29–30	Sup.
14	Magazine-cooling system	...	2,500	1½	880–1,140	Berth.	S.	32	Closed.
15	Magazines and stores, 33–39	...	5,000	3	615–810	Berth.	S.	33–34	Sup.
16	Firemen's washrooms, air casings, etc	...	5,000	3	615–810	Berth.	P.	59–60	Sup.
17	Firemen's washrooms, air casings, etc	...	5,000	3	616–810	Berth.	S.	59–60	Sup.
18	Midship magazines	...	5,000	3	615–810	Berth.	P.	80–81	Sup.
19	Midship magazines	...	5,000	3	615–810	Berth.	S.	80–81	Sup.
20	Quarters and stores, 76–95	...	5,000	3	616–810	Berth.	P.	86–87	Sup.
21	Quarters and stores, 76–95	...	5,000	3	615–810	Berth.	S.	86–87	Sup.
22	Magazines, aft	...	5,000	3	615–810	Berth.	P.	100–101	Sup.
23	Magazine-cooling system	...	2,500	1½	880–1,140	Berth.	P.	101–102	Closed.
24	Magazines, aft	...	5,000	3	615–810	Berth.	S.	104–105	Sup.
25	Quarters, stores and steering-gear room, aft 112	...	5.000	3	615–810	Berth.	C. L.	117–118	Sup.
26	Engine room	...	10,000	6½	440–575	U. P.	P.	87–88	Sup.
27	Engine room	...	10,000	6½	440–575	U. P.	S.	87–88	Sup.

U. T. = Under turret.

watertight to the highest practical point above this deck. All ducts passing through the magazines are galvanized-steel seamless tubing, made watertight. A natural exhaust duct, equal to the area of mechanical supply duct, has been provided for each shell room and small-arms magazine and located as far as practicable from supply duct. The upper ends of the ducts are carried up close to the gun deck and inside of the barbettes where practicable, the lower end of which is bell-mouthed and covered with wire mesh.

McCrery or equally effective adjustable elbows, fitted with butterfly dampers, have been used for supply terminals in all quarters and living spaces. In the quarters they are nickel-plated and in other places galvanized. The supply terminals in store rooms and magazines are fitted with adjustable elbows. All openings of these terminals are fitted with portable wire mesh. All other terminals are stationary.

Where any of the cowls interfere with the working of the guns or where necessary for cleaning decks they are made portable at the weather decks, and where such cowls are too high above the weather decks to be easily reached they are fitted with operating gear worked from weather decks. Where required, cowls under awning have been provided with wire-mesh screens.

The ventilation for the coal bunkers has been arranged with ducts of galvanized-steel tubing. Valves and dampers have been fitted for this system, and all openings into bunkers have been covered with wire mesh to keep coal out of ducts.

Especial attention has been paid to the ventilation arrangements provided to avoid the transmission of heat from the engine and boiler rooms to other parts of the vessel. In wake of the boiler rooms under protective deck and in the boiler-hatch casings air casings have been fitted and connected to ship's ventilation supply. The under side of protective deck within the engine rooms has been fitted with sheathing, with an air space allowed, and connected to the engine-room ventilation system for supply and to the engine-room hatches for exhaust.

For ventilating the two engine rooms there are two fans so arranged that one fan supplies the starboard engine room and the other the port. Branches have been led from the main to all working stations, platforms, air casing and corners of the rooms where required for efficient ventilation.

DOUBLE-BOTTOM COMPARTMENTS.

CAPACITIES OF DOUBLE-BOTTOM COMPARTMENTS.

Compartment.	Gallons.	Tons, F. W.	Tons, S. W.
A–92	10,085	37.4	38.5
A–93	12,261	45.5	46.8
A–94	22,871	84.9	87.3
A–95	20,517	76.2	78.3
A–96	14,751	54.7	56.3
A–97	23,183	86.1	88.5
A–98	12,917	47.9	49.3
A–99	12,917	47.9	49.3
B–76	6,969	25.8	26.6
B–77	6,969	25.8	26.6
B–78	5,659	21.0	21.6
B–79	5,659	21.0	21.6
B–80	5,868	21.7	22.4
B–81	5,868	21.7	22.4
B–82	7,361	27.2	28.1
B–83	7,361	27.2	28.1
B–84	7,361	27.2	28.1
B–85	7,361	27.2	28.1
B–96	11,056	41.0	42.2
B–97	11,056	41.0	42.2
B–98	13,833	51.4	52.8
B–99	13,833	51.4	52.8
D–97	19,072	70.8	72.8
D–98	19,963	74.1	76.2
D–99	24,128	89.6	92.1

RESERVE FEED TANKS IN INNER BOTTOM.

Compartment.	Gallons.	Tons, F. W.	Tons, S. W.
B–86	6,497	24.1	24.8
B–87	6,497	24.1	24.8
B–88	5,187	19.2	19.8
B–89	5,187	19.2	19.8
B–90	5,187	19.2	19.8
B–91	5,187	19.2	19.8
B–92	6,497	24.1	24.8
B–93	6,497	24.1	24.8
B–94	6,497	24.1	24.8
B–95	6,497	24.1	24.8

DRAINAGE SYSTEM.

The following is a general description of the drainage system, the features covered being as follow:

1. 15½-inch main drain.
2. 5½-inch secondary drain.
3. 5⅝-inch double-bottom drain.
4. 4⅝-inch forward-bilge drain.
5. 4⅝-inch aft-bilge drain.
6. 5-inch independent drain, firerooms.
7. 5-inch independent drain, engine rooms.

Main Drain.—This is 15½ inches diameter throughout its entire length. It is located on the starboard side close to the coal-bunker bulkhead, and extends from after end of forward fireroom to forward end of engine room, where it branches and runs athwartships, each branch connecting to a centrifugal pump.

A 15½-inch stop-check valve is located in each fire and engine room. All these 15½-inch stop-check valves are operated at the valves and also in deck plates on the middle protective deck.

In both engine rooms, at frame 84½, there is a 4½-inch connection between the main drain and the C. & R. manifold Nos. 5 and 6, located at frames 93–94. This manifold is connected to the fire and bilge pumps in the fire and engine rooms.

C. AND R. MANIFOLDS AND LOCATIONS OF SAME.

Manifold No.	Between frames.	Side of ship.	Compartment.
1	49 and 50	Center line.	B–1
2	58 and 59	Center line.	B–2
3	67 and 68	Center line.	B–3
4	72 and 73	Center line.	B–4
5	93 and 94	Starboard.	C–1
6	93 and 94	Port.	C–2
7	17 and 18	Starboard.	A–7
8	85 and 86	Starboard.	C–1
9	85 and 86	Port.	C–2
10	43 and 44	Starboard.	A–60 (Up. plat.)
11	95 and 96	Center line.	D–33 (Up. plat.)

STEAM ENGINEERING MANIFOLDS AND LOCATIONS OF SAME.

Manifold No.	Location.
18	Fireroom Number 1.
19	Fireroom Number 2.
20	Fireroom Number 3.
21	Fireroom Number 4.
28	Starboard engine room.
29	Port engine room.

Secondary Drain.—This is 5½ inches diameter throughout its entire length. It has connections to all fire and bilge pumps through the C. & R. and S. E. manifolds; also to the handy-billy pump manifold M–10, located on the upper platform, forward starboard frames 43 and 44, aft frames 95 and 96 port.

The forward connection is made directly to the auxiliary drain in dynamo room, frames 43 and 44, while the after connection is also made directly to the auxiliary drain in engine room, at frames 94 and 95.

The forward end of the auxiliary drain connects direct to the forward bilge-drain main, which also connects to the forward trimming tank, the drainage tank and inner bottom.

The after end of auxiliary drain connects directly to the

after bilge, after-drain main, which also connects with the after trimming tanks and inner bottom.

There is a 5½-inch connection from bilge well in each fireroom and in each engine room to the secondary drain.

Double-Bottom Drain.—The double-bottom drain is divided into three sections, each being 4⅝ inches in diameter.

The forward section runs aft on the port side from frame 44 to frame 61, and connects to the C. and R. manifolds Nos. 1 and 2 and drains the double-bottom spaces under the two forward firerooms and dynamo room.

The middle section extends from frame 67 to frame 73 on the port side and connects to C. and R. manifolds Nos. 3 and 4, and drains the double-bottom spaces under the two after firerooms.

The after inner-bottom drain consists of two sections, one starboard and one port, which extend from frame 86 to frame 93 and drain the inner-bottom spaces from frame 76 to frame 95.

The double-bottom compartments are flooded through the S. E. manifolds, connected to C. and R. manifolds operating the stop-check-lift valves in their respective manifolds.

Forward Bilge Drain.—It is 4⅝ inches in diameter and drains all double bottoms forward of bulkhead No. 39, also the drainage tank, bulkheads Nos. 23 and 24 and forward trimming tanks. It connects direct to the auxiliary drain between frames 43 and 44.

After Bilge Drain.—It is 4⅝ inches in diameter and drains all double-bottom compartments aft of bulkhead No. 95, and also after trimming tanks. It connects direct to auxiliary drain between frames Nos. 94 and 95 in starboard engine room.

Independent Drain.—These drains come under the cognizance of the Bureau of Steam Engineering. In the firerooms they are 5 inches in diameter and run from the bilge wells to the fire and bilge pumps. These drains are fitted in each fireroom.

The independent drains in each engine room are 5 inches

diameter and come under the cognizance of the Bureau of Steam Engineering. They are fitted in each engine room, and drain the bilge and crank-pit wells in the engine rooms. The draining of the shaft alley is to the engine-room bilges, from which there is a direct connection to the Bureau of Steam Engineering manifold, which is connected to the fire and bilge pumps.

TRIMMING TANKS, PUMPING AND DRAINAGE.

There are four trimming tanks, two forward and two aft. Those forward are flooded by 5-inch sea connections, frames 10 and 11, starboard side. A 4-inch pipe is led to each trimming tank, which is controlled by a 4-inch stop valve, operated at valve and on berth deck in deck plates, frames 10 and 11.

Those aft are flooded by a 5-inch sea connection, frames 112 and 113, starboard side. A 4-inch pipe is led to each trimming tank, which is controlled by a 4-inch stop valve, operated at valve on berth deck in deck plates, frames 112 and 113.

Draining.—The forward tanks are drained by a 3-inch pipe lead to manifold M–7, frames 17 and 18, and after tanks by a 3-inch pipe lead direct to the after-bilge suction, starboard side, frames 104 and 105. The extreme forward and after tanks are sluiced into adjacent compartments.

CAPACITIES OF TRIMMING TANKS.

Compartment.	Gallons.	Tons, F. W.	Tons, S. W.
A–1	20,982	77.9	80.1
A–2	20,488	76.1	78.2
D–7	23,581	87.6	90.0
D–8	5,664	21.0	21.4

MAGAZINE–FLOODING SYSTEM.

The following is a general description of the piping of the magazine-flooding systems:

There are three systems—one forward, one midship and one aft. Each system floods the magazine spaces on the lower

platform, the upper-platform systems being supplied by connections to fire main. In addition to sea floods all magazines on lower platform are fitted with sprinkling systems connected to the fire main.

Forward System.—It is composed of two separate systems, one on the starboard and one on the port side, each being supplied by an 8-inch sea connection, between frames 35 and 36, starboard and port sides.

Adjacent to the sea connections are 8-inch globe valves in hold, operated at valves and on berth deck in deck plates.

The systems are run below lower-platform beam, branches being led up through the deck to the valves in magazine spaces on lower platform.

Midship System.—It is composed of two systems, one on the starboard and one on the port side, supplied by an 8-inch sea connection starboard and a 7-inch sea connection on the port side, between frames 79 and 80 on the lower platform. Adjacent to the sea connections are globe valves, 8-inch on starboard and 7-inch on the port side, operated at valves and on berth deck in deck plates. The systems are run below the upper-platform beams, branches being dropped to the valves.

After System.—It is composed of two separate systems, one on starboard and one on port side, being supplied by an 8-inch valve on the port side and 7-inch on starboard side, frames 104 and 105. Adjacent to the sea connections are 8- and 7-inch globe valves, on lower platform, operated at valve on berth deck in deck plates.

The systems run below the upper-platform beams, branches being dropped to the valves in magazine spaces.

Upper-platform Systems.—These are connected to the fire main, and consist of sprinkling system only.

All magazine-flood and sprinkling valves are gate valves, operated at valve and on berth deck in deck plates.

All pipes are of copper, tinned.

All pipe sizes are internal.

Sprinkling Systems.—A sprinkling system is fitted in magazines and shell rooms; it consists of 2½-inch brass pipe di-

vided into smaller branches, the latter being perforated on the under side and fitted with 2½-inch cut-out valves in each magazine, operated at valve and on berth deck in deck plates. These are connected with the fire system, which serves the flooding systems on upper and lower platforms, all of which are shown on the plan of the sprinkling system for the forward midship and after magazines. All valves are brass gate valves.

DESCRIPTION OF FIRE MAIN.

A fire main of tinned copper is worked under the protective deck nearly the entire length of the ship. The fire main is divided into seven systems, viz: Starboard and port, forward and after, and forward, midship and after-magazine system, with cut-out valves for each system.

The starboard and port systems are each 6 inches in diameter and run outboard, starboard and port, below the protective deck throughout the length of the machinery spaces. The cross connection forward passes through the 12-inch handling room and the after cross connection through the after end of the engine rooms. This circuit is connected by a 5-inch riser to the fire and bilge pumps in each boiler room, and by two 5-inch risers to the fire and bilge pumps in each engine room. The distiller-circulating pumps in the evaporator room on the berth deck can also charge the fire main through a 6-inch connection to this circuit in the blower rooms on the upper platform outboard of each engine room.

In this circuit there are four valves, two in the forward cross connection and two in the after cross connection, and they are operated either at the valves or on the berth deck in deck plates. The other systems branch from this circuit and are taken from between the cut-out valves in the cross connections mentioned above, except the midship-magazine system, which is supplied by 5-inch branches from the starboard and port system.

Branches, varying from 2½ inches to 6 inches, depending on their uses, are taken from the two systems.

In general all fire plugs are 2½ inches in diameter.

The fire main is by-passed to the flushing main on gun deck, between frames 20 and 21, by a 6-inch gate valve.

All fire and bilge pumps are protected by 3-inch angle relief valves, set to discharge at 125 pounds pressure.

A 5-inch branch, containing gate valve, is taken from the after cross connection in starboard engine room, frames 94 and 95, and supplies the sprinkling system for after magazines. The midship magazine system is connected to both the starboard and port systems, by 5-inch branches with gate valves, at frame No. 69.

Cut-outs for these systems are operated at valve and on berth deck in deck plates.

Fire plugs Nos. 2 and 3, in port and starboard ammunition passage, lower platform, frame No. 38, and fire plug No. 11, frame No. 28, in upper platform and forward handling room, are supplied from the forward magazine system.

Fire plug No. 13, on upper platform, bulkhead No. 76, in 12-inch handling room, is supplied by the midship magazine system. Fire plugs Nos. 14 and 15, at bulkhead No. 95 and frame 105, respectively, in the 12-inch handling room on the upper platform, are supplied by the after magazine system.

Six-inch connections from the starboard and port systems, with cut-out valves operated at valve and at berth deck in deck plates, are led to coal bunkers, starboard and port, for flooding.

DECK DRAINS.

The officers' showers and baths, servants' wash rooms, general mess pantry, officers' and crew's galleys and crew's water closet on gun deck are drained by 2-inch pipe, and the firemen's wash room by 3-inch pipe, discharging overboard through ship's scupper.

The plumbing spaces on the main and forecastle decks are fitted with independent drain, and discharge through ship's scuppers.

FRESH-WATER SYSTEM.

A fresh-water main of galvanized wrought iron, in general of 2½ inches in diameter, runs the length of the ship, and is supplied either by the two electric fresh-water pumps or from two gravity tanks, one of 1,200 gallons capacity, located on the boat platform, and one 800-gallon tank on top of chart house.

Two and one-half-inch risers from the fresh-water main to forward and after gravity tanks serve as filling and discharge pipes for tanks at frames 57 and 58 and frames 46 and 47, where these tanks are located.

From the tank on the chart house this 2½-inch line runs forward on the port side on forecastle and main decks to a point between frames 12 and 13, where it passes down to the main, gun and berth decks to pump room.

From the tank on the boat platform this 2½-inch line passes down to berth deck, turns to starboard and runs forward to frame 23, where it passes down to forward protective deck and continuing forward to pump room.

The after 2½-inch line starts at frame 54 on berth deck, runs aft along starboard passage and through crew's space aft to frame 100, where it reduces to 1½-inch and continues aft to frame 104, turns to port, then forward to a point between frames 98 and 99, where it passes up to gun deck, and then divides into smaller branches, supplying plumbing spaces in that vicinity. From this 2½-inch line branches are taken supplying wash rooms, pantries, laundries, etc., on the various decks.

Discharges from pumps are 2½ inches in diameter. Cut-out valves are fitted near tanks in all filling and suction pipes.

Check valves are fitted where necessary.

Two and one-half-inch pipes, serving as air escape and overflow, are fitted to all tanks in hold ; 3-inch overflow pipes are fitted to the gravity tanks.

Tanks in hold are filled from deck on water boat through 2½-inch hose connections, starboard and port, at ship's side, under main deck, between frames 22 and 23.

CAPACITIES OF FRESH-WATER TANKS.

Comp't or tank.	Location.	Bet. frames.	Gals.	Tons.
A–3	Hold S.	9–13	4,614	17.1
A–4	Hold P.	9–13	4,614	17.1
A–5	Hold S.	13–17	7,358	27.3
A–6	Hold P.	13–17	7,358	27.3
Gravity tank, top of chart house,		45½–46½	800	3.0
Gravity tank, boat platform,		57–58	1,200	4.5

Filling pipes from ship's side, starboard and port, between frames 62 and 64, gun deck, to reserve-feed tanks, are by-passed on berth deck, frames 63 and 64 in boiler-hatch water main, and may be used for filling tanks in hold.

A 2-inch distiller main extends from frame 97 on berth deck along starboard side to a point between frames 23 and 24, by-passed down to forward protective deck, and there connects with the filling pipes to tanks. Branches from this distiller line are arranged to fill through two-way cocks the copper tanks located on the main various decks for firemen's, petty officers' and servants' wash-rooms and those in laundry, these being the only rooms supplied with tanks.

Two 3-H.P. electric pumps, having a capacity of 60 gallons per minute each, are provided for supplying the fresh-water main and filling the gravity tanks on boat platform and chart house from the tanks in hold. Both electric pumps are located on the forward protective deck in a pump room, starboard side, between frames 8 and 13.

One and one-half-inch relief valves on pump discharges protect the pumps and are set to relieve at 30 pounds pressure.

A 1½-inch hose valve on superstructure just above main deck, port side, frame 39, supplies steam launches with fresh water.

Branches from main supply all spaces requiring fresh water, such as lavatories, bath rooms, wash rooms, dispensary, operating room, pantries, galleys, bakery and laundry, also water butts and ice machine.

Three-quarter-inch faucets, for bucket use, are fitted in

seamen's head, firemen's wash room, and for stateroom use, in passage.

All fixtures are supplied direct from fresh-water main, except those in firemen's, petty officers' and servants' wash rooms.

All branches have cut-out valves at or near the main.

SALT-WATER SYSTEM.

A flushing main of copper, 6 inches in diameter, and tinned, extends along port side of berth deck, from frame 94 to frame 26, where it passes up to gun deck, running forward to frame 20, and is there by-passed to fire main.

Branches from this portion supply the servants', firemen's and petty officers' wash rooms, junior and warrant officers' and crew's waterclosets, and laundry, galley and general mess pantry, all on the gun deck, ash chutes, starboard and port, crew's wash room, ward room, officers' watercloset and showers on the main deck, and the captain's executive officers' bath rooms on the forecastle deck.

This flushing main is charged by the distiller circulating pumps in the evaporator room on berth deck, which are connected between frames 82 and 83, port side. The forward and after portions of the flushing main branch from this connection in crew's space on berth deck, and are fitted with cut-out valves at this junction.

A 6-inch globe valve, between frames 20 and 21, center line, on gun deck, permits charging the flushing main directly from the fire main.

The flushing system is also directly connected by 5-inch risers from the fire and bilge pumps in boiler rooms B–1 and B–2, and by a 6-inch riser from the fire and bilge pumps in the engine rooms.

A 3-inch relief valve in crew's head, port side, forward of bulkhead 12, set at 30 pounds, discharges into ship's scupper.

Three-quarter-inch faucets for bucket use are fitted in crew's and firemen's wash rooms.

All branches taken from flushing main have cut-out valves at or near the main.

No salt-water tanks are fitted on the ship.

Two valve manifolds on upper platform, one at bulkhead 44, starboard side, in passage, and the other on center line at bulkhead 95, have connections from sea and bilge, and are designed for use with the 6-inch handy-billy pumps.

In addition circulating water for ice machines is taken from a 3-inch connection at the dynamo-room injection sea valve.

The discharges overboard are through ship's scupper.

OFFICIAL TRIALS.

Standardization Trials.—The vessel was standardized over the measured-mile course off Rockland, Me., October 22 and 23, 1909. Nine runs were made on October 22d and eleven on October 23d.

Draught, forward, feet and inches	26–06¼
aft, feet and inches	27–11¼
mean	27–02
Corresponding displacement, tons	20,209

The data for standardization runs is recorded in Tables III and IV, and curve of revolutions and horsepower is shown on Plate.

During the high-speed runs on October 23d, from 10 A. M. to 12 M., the oil fuel was used in combination with coal. The oil-fuel apparatus worked satisfactorily. The following data of this test was obtained.

OIL-FUEL DATA.

Pressure of steam on oil heaters, pounds	178 to 274
discharge to burners, pounds	115 to 147
Temperature, oil-settling tanks, degrees Fahrenheit	68
oil discharge from heaters, degrees Fahrenheit	292 to 300
Double strokes of oil-service pump, No. 1, F. R.	20
2, F. R.	35
3, F. R.	62
4, F. R.	19

Table III.—U. S. S. *DELAWARE.*

DATA FROM PROGRESSIVE RUNS OVER MEASURED-MILE COURSE OFF ROCKLAND, ME., OCTOBER 22-23, 1909.

Number of run.	Average speed for group of runs.	Average for group of runs.	Revs. of main engines. Per minute. Starboard.	Port.	Mean.	Speed, in knots per hour.	Indicated horsepower. Starboard engine. H.P. cylinder.	I.P. cylinder.	F.L.P. cylinder.	A.L.P. cylinder.	Total.	Port engine. H.P. cylinder.	I.P. cylinder.	F.L.P. cylinder.	A.L.P. cylinder.	Total.	Total of both main engines.	Total, all auxiliaries.	Total for all machinery.
1			53.74	54.24	53.99	10.169	268	296	176	206	946	270	332	108	175	885	1,831	335	2,166
2	9.784	53.9	54.25	53.94	54.09	9.685	210	282	148	178	818	261	330	103	153	847	1,665	346	2,011
3			53.28	53.54	53.41	9.597	236	277	152	175	840	296	652	155	164	967	1,807	349	2,156
4			70.28	70.37	70.32	12.83	695	572	392	404	2,063	621	646	366	423	2,056	4,119	358	4,477
5	12.409	68.734	68.04	67.96	68.01	12.08	640	563	354	372	1,929	559	643	649	401	1,952	3,881	360	4,241
6			69.11	68.08	68.60	12.645	669	571	348	378	1,966	545	573	298	372	1,788	3,754	369	4,123
7			89.61	88.01	88.81	15.537	1,451	1,248	931	858	4,488	1,191	1,292	751	833	3,987	8,475	357	8,832
8	15.976	89.796	91.32	90.06	90.69	16.484	1,498	1,264	837	894	4,493	1,256	1,295	837	862	4,250	8,743	418	9,161
9			89.81	88.16	88.98	15.398	1,429	1,265	835	894	4,423	1,199	1,194	723	824	3,940	8,363	402	8,765
10			109.97	109.3	109.63	19.726	2,633	2,239	1,684	1,654	8,210	2,158	2,158	1,614	1,649	7,579	15,789	614	16,403
11	19.337	109.299	109.16	108.75	108.95	18.905	2,568	2,206	1,635	1,612	8,021	2,027	1,964	1,641	1,546	7,178	15,199	583	15,782
12			110.24	109.04	109.64	19.813	2,715	2,329	1,748	1,748	8,540	2,258	2,136	1,640	1,669	7,703	16,243	615	16,858
13			119.58	119.47	119.53	20.259	3,442	3,167	2,256	2,289	11,154	3,134	3,108	2,287	2,385	10,914	22,068	791	22,859
14	20.808	120.298	120.67	120.46	120.56	21.289	3,373	3,140	2,375	2,323	11,211	3,193	3,171	2,404	2,437	11,205	22,416	815	23,231
15			120.58	120.47	120.52	20.393	3,137	3,212	2,393	2,361	11,103	3,127	3,097	2,371	2,437	11,032	22,135	824	22,959
16			126.13	126.13	126.18	21.089	3,491	3,862	2,897	2,863	13,113	3,466	3,803	2,862	2,966	13,097	26,210	997	27,207
17			124.87	125.64	125.25	21.489	3,905	3,727	2,629	2,595	12,856	3,469	3,661	2,707	2,817	12,654	25,510	886	26,396
18	21.44	126.928	127.13	126.97	127.05	21.286	4,113	3,931	2,850	2,885	13,779	3,629	3,419	2,777	2,951	12,776	26,555	849	27,404
19			128.45	128.85	128.65	21.664	4,390	4,326	2,810	2,845	14,371	3,700	3,944	2,783	2,889	13,316	27,687	947	28,634
20			126.99	127.64	127.32	21.582	4,218	4,141	2,743	2,722	13,824	3,560	3,750	2,806	2,966	13,082	26,906	936	27,842

TABLE IV—U. S. S. *DELAWARE.*

DATA FROM PROGRESSIVE RUNS OVER MEASURED-MILE COURSE, OFF ROCKLAND, ME., OCTOBER 22-23, 1909.

TABLE IV—U. S. S. *DELAWARE*.

DATA FROM PROGRESSIVE RUNS OVER MEASURED-MILE COURSE, OFF ROCKLAND, ME., OCTOBER 22-23, 1909.

Number of run.	Which engine.	Steam pressures.					Mean effective pressures.					Vacuum, in inches.	Auxiliary exhaust, gauge.	Air pressure, in inches of water.	Cut-off in cyls. from beginning.				Temperatures.								Revolutions or double strokes.						
		In boilers, above atmosphere.	At engine, above atmosphere.	H.P. steam chest, absolute.	In I.P. receiver, absolute.	In L.P. receiver, absolute.	H.P. cylinder.	I.P. cylinder.	Forward L.P. cylinder.	After L.P. cylinder.	Referred to L.P. cylinder.				High pressure.	Intermediate press.	Forward L.P.	After L.P.	Outside air.	Engine-room working platform.	Fireroom working level.	Injection.	Discharge.	Hotwell.	Feed.	Smoke pipes, forward and aft.	Bilge pumps.	Forced lubrication, av. of 3 pumps.	Main feed pumps.	Main feed pumps.	Circulating pumps.	Air pumps.	Fireroom blowers.
1	S. P.	275	272	40	12.5	6.5	18.0	9.0	3.0	3 5	8.05	27.3	4.0	-	.86	.80	.56	.56	51	86	80–108	51.0	59	188	132	260	13 0	26	10.8	10.0	156	14.0	-
2	S. P.	274	45	15.0	7.5	18 0	10.0	2.0	2.95	7-55	28.5	5.0	-	-	.86	.80	.62	.62	51	86	80–108	51.0	55	141	133	300			7.0	7.0	140	18.0	-
3	S. P.	265	262	39	12.5	6.3	14.0	8.5	2.5	3.0	6 9	27.5	4.0	-	.86	.80	.62	.62	51	86	80–108	50.0	59	180	160	255	13 0	24	14.0	14 0	144	17.0	-
4	S. P.	255	45	15.0	7.2	20.0	17.5	10.0	1.75	2.6	7.18	28 4	5.0	-	.86	.80	.62	.62	51	87	75–107	51.0	59	129	120	340	13.0	26	7.0	7.0	140	19.0	-
5	S. P.	285	288	42	12.5	6 5	16 0	8 5	2.6	3.0	7.2	27.6	4.0	-	.86	.80	.62	.62	51	87	75–107	50.0	59	171	142	180			14.0	14.0	144	15.0	-
6	S. P.	269	285	45	15.0	7.2	20.0	10.75	2.65	2 8	8.26	28.5	5 0	-	.86	.80	.62	.62	51	87	76–109	51.0	59	129	120	340	13.0	26	7.5	8.5	140	14.0	-
7	S. P.	278	268	64	27.0	10.5	35.75	13 3	5.1	5.25	13.42	27.7	4 0	-	.86	.80	.62	.62	52	87	76–109	50.0	60	155	148	260			15.0	15 0	154	16.0	-
8	S. P.	282	278	71	29.0	14.2	32 0	15.0	4.75	5 5	13.36	28.4	5.0	-	.86	.80	.62	.62	52	87	78–109	51 0	61	110	120	300	12.5	25	7.0	8.0	140	13.5	-
9	S. P.	270	274	63	27.0	10.0	34.0	13.5	4.75	5.0	12.95	28.4	4 0	-	.86	.80	.62	.62	52	87	78–109	51 0	60	138	120	220			17.0	17.0	146	14.0	-
10	S. P.	278	270	65	30.0	14 2	29 8	15.45	4.7	5.4	13.18	28.4	5.0	-	.86	.80	.62	.62	52	87	80–111	50.0	60	108	105	320	13.8	31	7.5	8.0	135	16.0	-
11	S. P.	278	279	62	27.0	10.0	35 0	13.5	4.6	5.0	13.0	28.6	3.5	-	.86	.80	.62	.62	52	87	80–111	50.5	60	120	112	280			17.0	17.0	154	20.0	-
12	S. P.	278	278	62	27 0	13.8	29.0	13.75	4.0	5.0	12 01	28.4	5.0	-	.86	.80	.62	.62	52	87	80–111	50.5	60	108	98	360	13.0	25	7.5	8.0	140	14.5	-
13	S. P.	278	277	112	47.0	17.0	58.5	22.75	9.6	8.75	22.85	28.5	4.0	-	.86	.80	.62	.62	52	87	90–111	51.0	70	115	108	245			15.0	15.0	150	19.0	-
14	S. P.	282	275	102	43.0	17 0	49.0	22.5	7 8	8.65	20.71	28.4	5 0	-	.86	.80	.62	.62	52	88	90–111	51.0	74	93	90	360	12 0	25	7.5	8.0	140	15.0	-
15	S. P.	282	286	110	45.0	17.5	59 25	22.6	8 38	8 95	22 5	28.4	4.0	-	.86	.80	.62	.62	52	88	87–113	51.0	75	105	95	200			18 0	18.0	156	19.0	-
16	S. P.	270	278	123	43.0	17.0	50.5	23.5	8.5	8.75	21.57	28.4	5.0	-	.86	.80	.62	.62	54	88	87–116	51.0	74	85	84	370	13 0	24	8.0	8.5	138	16 0	-
17	S. P.	270	268	110	45.0	17 5	57.5	23.0	8.5	9.1	22.52	28.7	3.5	-	.86	.80	.62	.62	54	88	87–116	51.0	85	98	87	280	12.0	33	18.0	18.0	156	19 0	-
18	S. P.	278	267	114	42.0	16 8	49 25	22 13	7.5	8 55	20.44	28.1	5.0	-	.86	.80	.62	.62	53	88	92–98	51.0	74	90	170	380			14 0	13.5	140	14.0	-
19	S. P.	270	274	178	72.0	27.5	86 5	33 25	14.0	13.75	34.13	27.7	6.0	0 45	.86	.80	.62	.62	53	86	92–98	51.0	100	100	160	280	-	40	20.0	20.0	180	18.0	-
20	S. P.	278	279	170	69.0	26.7	71.5	32.25	13.5	13 8	31.71	27.5	5.0	-	.86	.80	.62	.62	53	88	92–98	52.0	100	100	150	445	-	40	18.0	20.0	145	18.0	-
21	S. P.	289	288	178	72.0	27.5	85 0	33.0	13.7	13 5	33.59	27 9	10.0	0.45	.86	.80	.62	.62	53	87	90–109	51.0	90	100	160	320	-	41	18.0	18.0	190	16.0	550
22	S. P.	300	283	162	70.0	27.0	67 5	29.5	13.8	13 0	30.18	27.6	7.0	-	.86	.80	.62	.62	53	88	90–109	52.0	94	96	150	420	-	41	18.0	20.0	150	18.5	-
23	S. P.	300	300	175	71.0	27.0	89 0	34.5	14.5	14.5	35.42	28.0	4.0	0.4	.86	.80	.62	.62	52	88	90–115	51.0	90	97	150	340	-	43	19.0	19 0	185	15.0	550
24	S. P.	270	297	176	68.0	26 5	75.0	32.0	13.75	14.0	32.3	27.2	10 0	-	.86	.80	.62	.62	52	88	90–115	52.0	95	98	148	390	-	43	16.0	20.0	152	18.0	-
25	S. P.	270	260	225	89.0	33.0	104.0	43.25	17 25	17.5	42.65	27.0	4.5	1.0	.86	.80	.62	.62	52	91	90–97	51.0	105	105	150	400	-	37	28.0	28.0	204	17.0	556
26	S. P.	305	260	225	90.0	33.0	95.0	42.5	17.5	18 25	41.75	27 0	5.0	-	.86	.80	.62	.62	52	88	90–97	52.0	102	105	146	530	-	37	21.0	18.0	180	17.0	-
27	S. P.	296	296	227	91 0	34.0	101.0	42.5	18 0	19.6	42 48	27.1	4.0	1.3	.86	.80	.58	.58	50	90	88–95	51.0	110	110	160	380	-	54	18.0	18.0	216	15.0	570
28	S. P.	290	290	229	90 0	33.7	96.0	43.0	18.25	18.5	42.52	26 5	5.0	-	.86	.80	.58	.58	50	90	88–95	52.0	100	108	150	560	-	54	22.0	22.0	165	18.0	-
29	S. P.	296	287	227	92.0	34.5	94.0	43.5	18.15	19.9	42.1	27.4	4.0	1.1	.86	.80	.58	.58	50	91	80 93	51.0	100	100	160	340	-	41	21.0	21.0	215	20.0	570
30	S. P.	294	280	230	90 0	34 0	94.0	42.0	18.0	18 5	41.87	27.2	5.0	-	.86	.80	.58	.58	50	92	80 93	52.0	100	105	146	530	-	41	21.0	21.0	165	17.0	-
31	S. P.	285	283	252	106.0	39 5	100.0	50.0	21.0	20.75	47 52	27.0	5.0	0.9	.86	.80	.58	.58	50	90	80–105	51.0	110	110	150	360	-	38	20.0	20.0	216	18.0	575
32	S. P.	285	280	247	108.0	39.0	99.5	49.25	20 75	21.5	47 46	26.7	5.0	-	.86	.80	.62	.62	52	92	80–105	52.0	103	115	158	580	-	38	25.0	25.0	178	18.0	-
33	S. P.	275	272	245	100.0	37.0	113 0	48.75	19.25	19 0	47.07	26.5	5.0	1.2	.86	.80	.62	.62	52	89	82–109	51.0	110	115	150	420	-	45	24.0	24.0	204	18.0	575
34	S. P.	286	274	246	103.0	37.0	100.0	47.6	19.7	20.5	46.05	25 1	6.0	-	.86	.80	.62	.62	52	93	82–108	52.0	108	117	155	550	-	45	22.0	22.0	220	19.0	-
35	S. P.	286	280	254	107.0	38.5	116.9	50.5	20.5	20.75	49.55	26.4	5.0	1.3	.86	.80	.62	.62	54	92	82–108	51.0	110	120	160	410	-	41	18.0	18.0	210	19.0	605
36	S. P.	305	280	250	107.0	38.0	103 5	44.0	20.0	21 25	46 01	25.7	7.0	-	.86	.80	.62	.62	54	93	82–108	52.0	105	115	165	570	-	41	20.0	28.0	200	18.0	-
37	S. P.	305	288	268	109.0	38.0	123.5	55.0	20.0	20 25	51.15	27.0	6.0	1.4	.86	.80	.62	.62	56	90	82–104	51.0	110	120	170	420	-	61	20.0	20.0	230	20.0	630
38	S. P.	305	282	260	113.0	39.0	124.0	50.0	19.75	20.5	47.25	24.8	8.0	-	.86	.80	.62	.62	56	92	82–104	52.0	112	115	152	595	-	61	24.0	24.0	240	20.0	-
39	S. P.	287	270	252	107 0	37.0	120.0	53.25	19.75	19 6	49.79	-	7.0	1.4	.86	.80	.62	.62	56	90	82–102	51.0	105	110	165	430	-	47	22 0	22.0	220	20.0	620
40	S. P.		265	250	111.0	38.0	101.0	48.0	20.1	21.25	46.75	25.6	8.5	-							82–102	52.0	105	112	140	600	-	47	20.0	26.0	240	19.0	

POWER OF AUXILIARIES.

The various auxiliaries were tested and indicated previous to the trials, and the results plotted on the accompanying curves of revolutions—horsepower.

OFFICIAL FOUR-HOURS' TRIAL.

On the evening of October 23, 1909, the *Delaware* got under way and went to sea for the four-hours' official trial.

The draught and displacement on the trial were—

Draught, forward, feet and inches	25-11⅞
aft, feet and inches	27-09⅛
mean, feet and inches	27-00½
Corresponding displacement, tons	20,098.5

The trial commenced at 8.30 P. M., October 23, 1909.

PERFORMANCE—FOUR-HOURS' OFFICIAL TRIAL.

Steam Pressures. (*Average of one-half hourly observations.*)

	Starboard.	*Port.*
Mean steam pressure at boilers, pounds	300.0	
Max. steam pressure of engines, pounds, H.P. (chest)	285.0	285.0
Mean steam pressure at engines, pounds	285.5	
H.P. steam chest gauge, lbs.	253.0	253.0
1st receiver (absolute), pounds,	105.0	110.0
2d receiver (absolute), pounds,	33.05	40.48
Vacuum in condensers, inches of mercury, mean	26.31	

Temperatures. (*Average of one-half hourly observations.*)

Injection, degrees	50.0	50.0
Discharge, degrees	109.0	100.0
Hotwell, degrees	109.0	100.0
Feed water, degrees	147.0	
Outside air, degrees	50.7	
Engine room, working platform, degrees	86.0	86.4
Firerooms, working level, degrees	88.5	
Smoke stacks, degrees, 400–500 average	491.0	
Superheat at engines, degrees	54.6	68.6
I.P. receiver, degrees	01.0	15.0

Revolutions, or double strokes, per minute. (*Average of one-half hourly observations.*)

Average revolutions, main engines, per minute	128.15	128.59
Mean revolutions, both engines, per minute	128.388	
Pumps, main air	18.4	21.9

	Starboard.	*Port.*
Pumps, circulating	222.0	214.0
feed, d.s., per minute	20.3	26.4
fire and bilge	12.0	51.4
auxiliary condenser, air and circulating	28.6	29.2
forced lubricating	43.0	44.7
Blower engines	668.0	
Speed of ship, in knots per hour	21.56	
Slip of propeller, in per cent. of its own speed, based on mean pitch	13.68	13.99
Air pressure in firerooms, in inches of water, mean	1.86	

Mean effective pressures in cylinders, in pounds per square inch. (*Averages of cards taken at half-hourly periods.*)

Main engines, H.P. cylinder	117.91	105.06
I.P. cylinder	52.53	52.84
F.L.P. cylinder	25.67	20.44
A.L.P. cylinder	19.96	22.18
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	52.43	49.47

Indicated Horsepower.

Main engines, H.P. cylinder	4,187.0	3,725.0
I.P. cylinder	4,128.0	4,153.0
F.L.P. cylinder	3,601.0	2,870.0
A.L.P. cylinder	2,800.0	3,144.0
total	14,716.0	13,862.0
Collective H.P. of both main engines	28,578.0	
Air pumps, main	4.96	5.5
Circulating pumps, main	120.0	108.0
Feed pumps, main, inboard	45.5	60.0
outboard	45.5	58.0
Dynamo condenser, air pump	2.06	
circulating pump	5.08	
Fire and bilge pumps	.5	13.6
Forced-draft blowers, electrically driven		
Dynamo engines	443.6	
Total, all auxiliaries	934.09	
Collective I.H.P. main engines, air, circulating and feed pumps	29,025.0	
Collective main and auxiliary engines in operation	29,512.0	

Coal.

Kind and quality used on trial.....New River, excellent hand-picked.
Pounds per hour, main and auxiliary engines, during trial..... 53,944.0

Deduced Data.

I.H.P. (total) per square foot of grate surface	20.509
Pounds of coal, per knot run	2,505.0
I.H.P. per hour, main engines only	1.888
all machinery in operation	1.827
square foot of grate surface, per hour	37.487
Cooling surface (main condenser), square feet per I.H.P., main engines only	.825
Heating surface, square feet per I.H.P. (total)	1.889

Water Consumption.

Pounds of water, per hour, total	422,931.0
main engines, per I.H.P., per hour	13.38
auxiliaries, per I.H.P. per hour	42.36
all purposes, per I.H.P. per hour	14.32

FORCED-LUBRICATION DATA ON FOUR-HOURS' FULL-SPEED RUN.

The oil pressures at pumps, starboard, 31.4 pounds; port, 31.7 pounds.

The pressure at bearings varied from 28.2 pounds at forward bearing to 24.8 pounds at after bearing, starboard engine, and 31.7 and 27.4 pounds for port engine.

The oil pressure at starboard crossheads, 2 to 4 pounds; port, 1 to 6 pounds.

The temperature in oil tanks, 102 to 108 degrees Fahrenheit, starboard; 99 to 105 degrees Fahrenheit, port.

Temperature at bearings varied from 92.2 degrees Fahrenheit at forward to 99.2 degrees Fahrenheit at after bearing for starboard engine, and 96.2 degrees Fahrenheit and 97.7 degrees Fahrenheit for port engine.

The forced-lubrication system operated with entire satisfaction on all of the trials and undoubtedly it served to add several revolutions per minute by reason of the reduced friction obtained.

TWENTY-FOUR HOURS' TRIAL AT 12 KNOTS.

This trial was held October 24th and 25th. Mean draught on trial, 26 feet $11\frac{1}{6}$ inches. Displacement, 20,061 tons. During trial there were light southeast breezes and moderate small to rough sea from southeast.

PERFORMANCE, TWENTY-FOUR HOURS' OFFICIAL TRIAL, AT 12 KNOTS.

Steam pressures. (*Average of hourly observations.*)

	Starboard.	*Port.*
Maximum steam pressure of boilers, pounds	271.0	
Mean steam pressure of boilers, pounds	252.0	
Maximum steam pressure of engines, lbs., H.P. (chest),	285.0	275.0
Mean steam pressure at engines, pounds	252.0	252.0
H.P. steam chest gauge, lbs	68.3	67.0
1st receiver (absolute), pounds	34.6	36.1
2d receiver (absolute), pounds	14.49	14.88
Vacuum in condensers, inches of mercury, mean	27.72	28.216

Temperatures. (*Average of hourly observations.*)

Injection, degrees	56.8	57.5
Discharge, degrees	69.5	75.1
Hotwell, degrees	92.2	89.1
Feed water, degrees	105.0	
Outside air, degrees	57.6	
Engine room, working platform, degrees	91.2	91.2
Firerooms, working level, degrees	108.0	
Smoke stacks, average, degrees	260–410	

Revolutions, or double strokes, per minute. (*Average of hourly observations.*)

Average revolutions, main engines, per minute	67.98	67.93
Mean revolutions, both engines, per minute	67.955	
Pumps, main air	19.2	22.8
circulating	135.0	143.0
feed, d.s., per minute	12.0	11.2
fire and bilge	26.56	9.68
auxiliary condenser, air and circulating	46.9	46.6
Blower engines	Natural draft.	
Speed of ship, in knots per hour	12.24	
Slip of propeller, in per cent. of its own speed, based on mean pitch	7.585	
Air pressure in firerooms, in inches of water, mean	Natural draft.	

Mean Effective Pressures in Cylinders, in pounds per square inch.

Main engines, H.P. cylinder	30.37	23.50
I.P. cylinder	14.97	13.82
F.L.P. cylinder	5.84	5.27
A.L.P. cylinder	5.74	5.91
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	13.82	12.86

	Starboard.	*Port.*
Cut-off, in decimals of stroke, H.P.	.86	
I.P.	.70	
L.P.	.38	

Indicated Horsepower.

Main engines, H.P. cylinder	570.0	443.0
I.P. cylinder	621.0	578.0
F.L.P. cylinder	433.0	394.0
A.L.P. cylinder	425.0	441.0
total	2,049.0	1,856.0
Collective H.P. of both main engines	3,905.0	
Air pumps, main	10.65	
Circulating pumps, main	60.2	
Feed pumps, main	52.1	
Dynamo condenser, air pumps	2.45	
Fire and bilge pumps	4.0	
Dynamo engines	154.3	
Total, all auxiliaries	341.0	
Collective I.H.P., main engines, air, circulating and feed pumps	4,028.0	
Collective main and auxiliary engines in operation	4,246.0	

Coal.

Kind and quality used on trial	New River, very good.
Pounds, per hour, main and auxiliary engines, during trial	10,439.0

Deduced Data.

I.H.P. (total) per square foot of grate surface	5.9
Pounds of coal per I.H.P. per hour, collective, main engines only,	2.673
all machinery in operation	2.458
square foot of grate surface, per hour	14.505
Cooling surface (main condenser), square feet per I.H.P., main engines only	6.038
Heating surface, square feet per I.H.P. (total)	6.565

Water Consumption.

Pounds of water per hour, total	83,463.0
main engines, per I.H.P. per hour	15.12
auxiliaries, per I.H.P. per hour	63.75
all purposes, per I.H.P. per hour	19.37

TWENTY-FOUR HOURS' TRIAL AT 19 KNOTS.

This trial was held October 25th and 26th. Mean draught on trial, 26 feet $9\frac{7}{8}$ inches. Displacement, 19,979 tons. There was a moderate sea with fresh N.N.W. breezes.

PERFORMANCE, TWENTY-FOUR HOURS' OFFICIAL TRIAL, AT 19 KNOTS.

Steam pressures. (*Average of hourly observations.*)

	Starboard.	*Port.*
Maximum steam pressure of boilers, pounds	300.0	
Mean steam pressure at boilers, pounds	279.4	
Maximum steam pressure of engines, lbs., H.P. (chest),	295.0	290.0
Mean steam pressure at engines, pounds	273.0	271.0
H.P. steam chest gauge, lbs	176.3	178.5
1st receiver (absolute), pounds	82.5	83.8
2d receiver (absolute), pounds	35.72	51.35
Vacuum in condensers, inches of mercury, mean	26.88	27.45

Temperatures. (*Average of hourly observations.*)

Injection, degrees	62.3	61.1
Discharge, degrees	102.9	91.0
Hotwell, degrees	96.8	82.6
Feed water, degrees	96.8	82.6
Outside air, degrees	51.2	
Smoke stacks, average, degrees	420–660	
Degrees superheat at boilers	56.4	64.9

Revolutions, or double strokes, per minute. (*Average of hourly observations.*)

Average revolutions, main engines, per minute	108.8	108.4
Mean revolutions, both engines, per minute	108.6	
Pumps, main air	20.0	22.4
circulating	205.6	200.0
feed, d.s., per minute, inboard	14.88	18.6
outboard	16.68	19.0
fire and bilge	11.4	25.2
auxiliary condenser, air and circulating	46.9	56.6
Blower engines	570.0	
Speed of ship, in knots per hour	19.217	
Slip of propeller, in per cent. of its own speed, based on mean pitch	9.277	
Air pressure in firerooms, in inches of water, mean	.448	

Mean Effective Pressures in Cylinders, in pounds per square inch.

Main engines, H.P. cylinder	72.8	62.8
I.P. cylinder	33.68	33.21
F.L.P. cylinder	20.5	15.49
A.L.P. cylinder	15.95	16.18
Mean equivalent pressure, in pounds per square inch, referred to combined area of L.P. pistons	36.87	33.05
Cut-off, in decimals of stroke, H.P.	.86	
I.P.	.70	
L.P.	.38	

Indicated Horsepower.

	Starboard.	Port.
Main engines, H.P. cylinder	2,195.0	1,872.0
I.P. cylinder	2,247.0	2,200.0
F.L.P. cylinder	2,442.0	1,832.0
A.L.P. cylinder	1,901.0	1,914.0
total	8,784.0	7,818.0
Collective H.P. of both main engines	16,602.0	
Air pumps, main	5.2	5.5
Circulating pumps, main	96.0	89.0
Feed pumps, main, inboard	33.5	41.8
outboard	37.5	45.0
Dynamo condenser, air pump	2.4	
Fire and bilge pumps	2.5	
Forced-draft blowers	Electrically driven.	
Dynamo engines	308.6	
Total, all auxiliaries	733.1	
Collective I.H.P., main engines, air, circulating and feed pumps	16,797.7	
Collective main and auxiliary engines in operation	17,335.0	

Coal.

Kind and quality used on trial	New River, very good.
Pounds, per hour, main and auxiliary engines, during trial	29,807.0

Deduced Data.

I.H.P. (total) per square foot of grate surface	12.05
Pounds of coal per I.H.P. per hour, main engines only	1.795
all machinery in operation	1.719
square foot of grate surface, per hour	20.92
Cooling surface (main condenser) square feet per I.H.P., main engines only	1.42
Heating surface, square feet per I.H.P. (total)	3.57

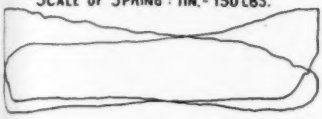
Water Consumption.

Pounds of water per hour, total	243,810.0
main engines, per I.H.P. per hour	12.7
auxiliaries, per I.H.P. per hour	40.72
all purposes, per I.H.P. per hour	13.91

NOTE.—The photo of *Delaware*, frontispiece of this number, was furnished by J. Abbe, of Newport News, Va.

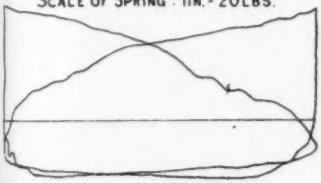
STARBOARD ENGINE.

H. P. CYLINDER.
M.E.P. 118.50.
I.H.P. 4216.
SCALE OF SPRING: 1IN. = 150 LBS.

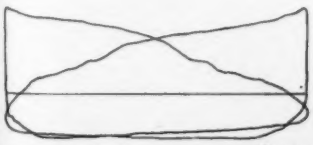


I. P. CYLINDER
M.E.P. 54.50.
I.H.P. 4290.
SCALE OF SPRING: 1IN. = 60 LBS.

F. L. P. CYLINDER.
M.E.P. 27.25.
I.H.P. 3831.
SCALE OF SPRING: 1IN. = 20 LBS.

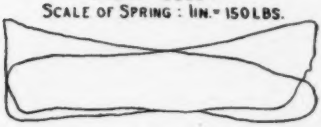


A. L. P. CYLINDER.
M.E.P. 20.40
I.H.P. 2868.
SCALE OF SPRING: 1IN. = 20 LBS.



PORT ENGINE.

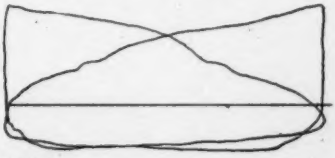
H. P. CYLINDER.
M.E.P. 110.09.
I.H.P. 3895.
SCALE OF SPRING: 1IN. = 150 LBS.



I. P. CYLINDER.
M.E.P. 54.50
I.H.P. 4278
SCALE OF SPRING: 1IN. = 60 LBS.

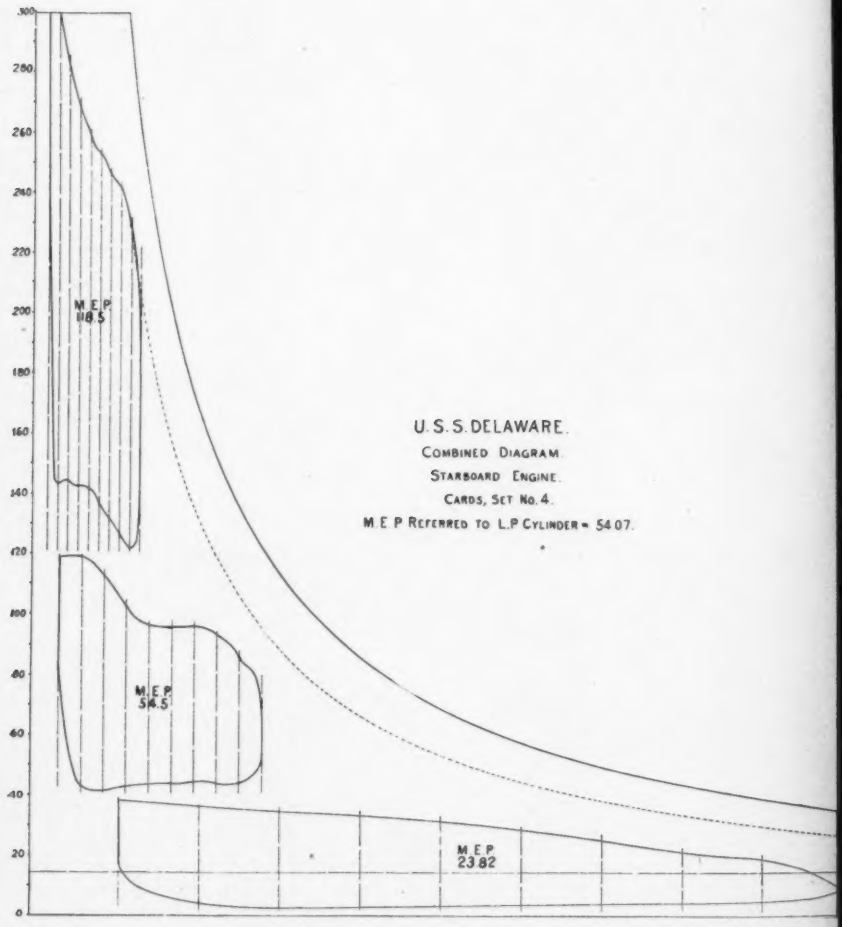
F. L. P. CYLINDER.
M.E.P. 21.75.
I.H.P. 3050.
SCALE OF SPRING: 1IN. = 20 LBS.

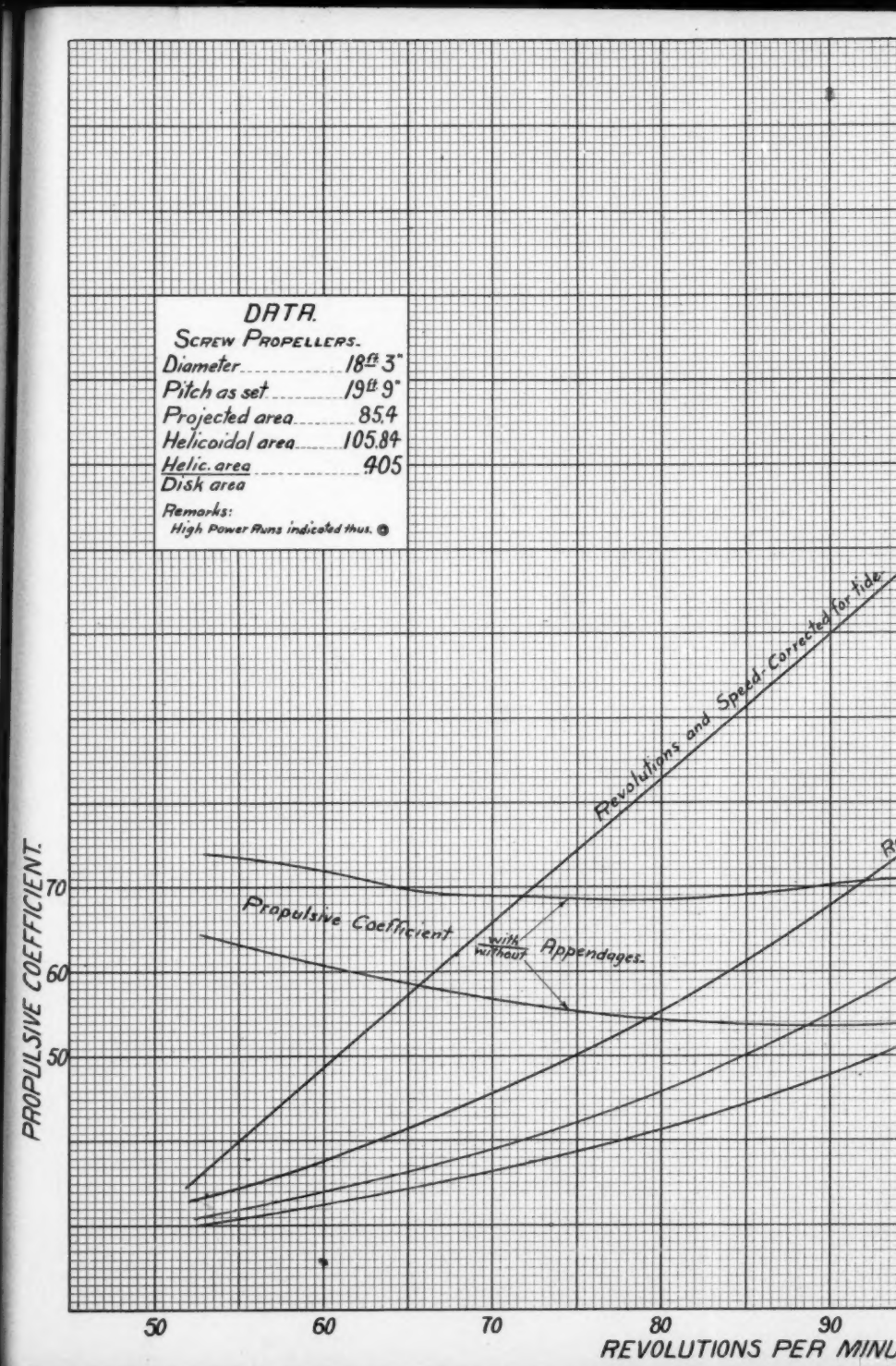
A. L. P. CYLINDER
M.E.P. 23.25
I.H.P. 3260.
SCALE OF SPRING: 1IN. = 20 LBS

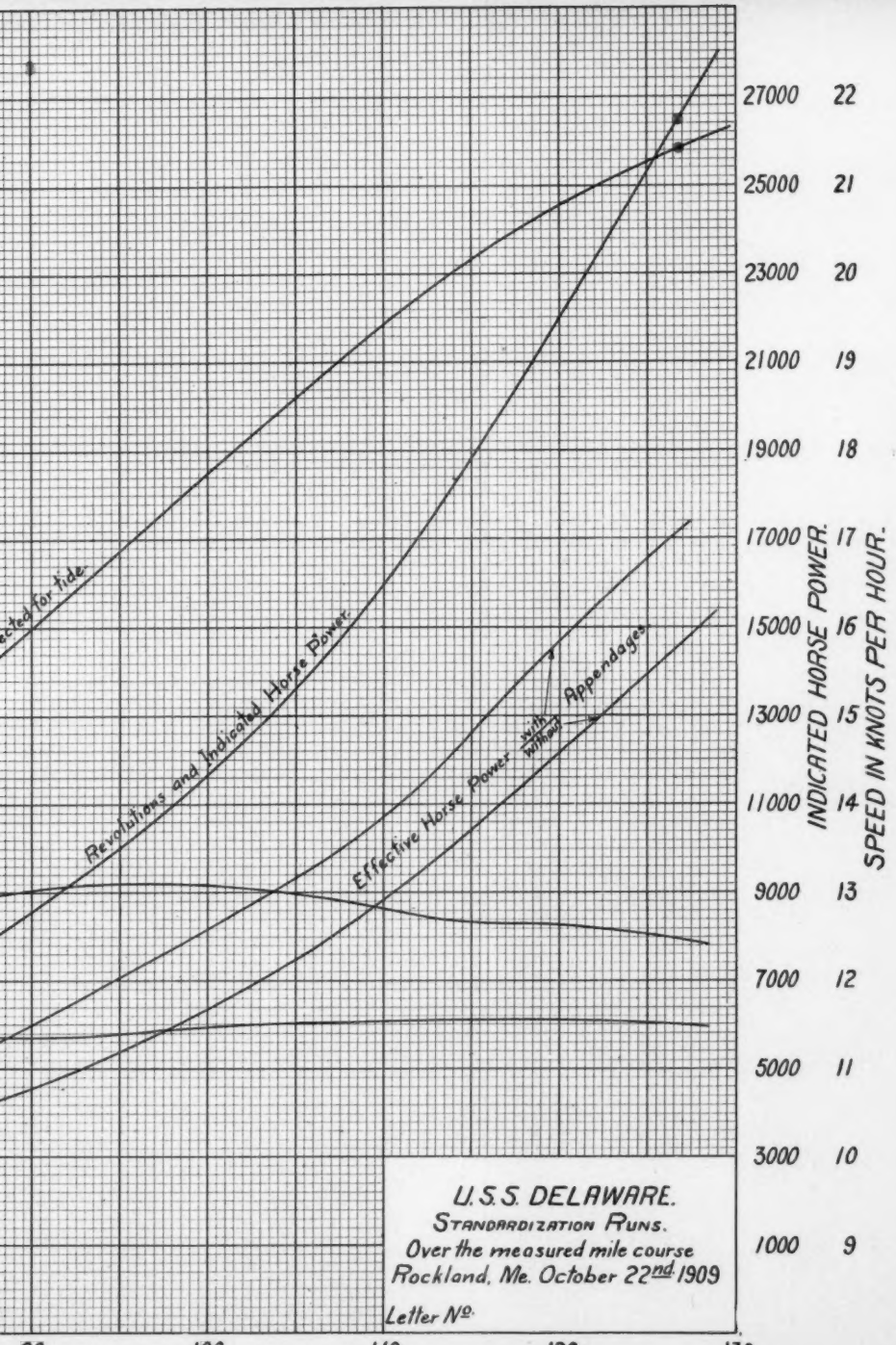


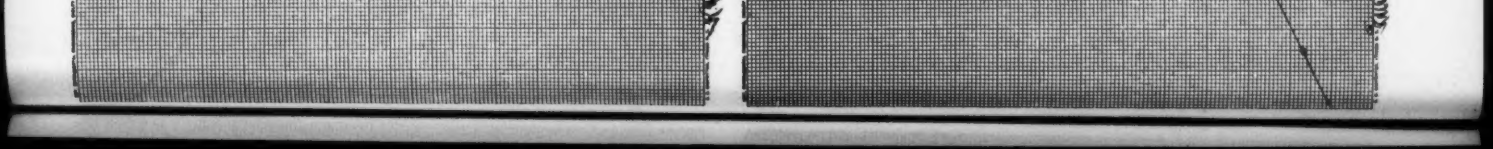
HALF SIZE OF ACTUAL DIAGRAMS.

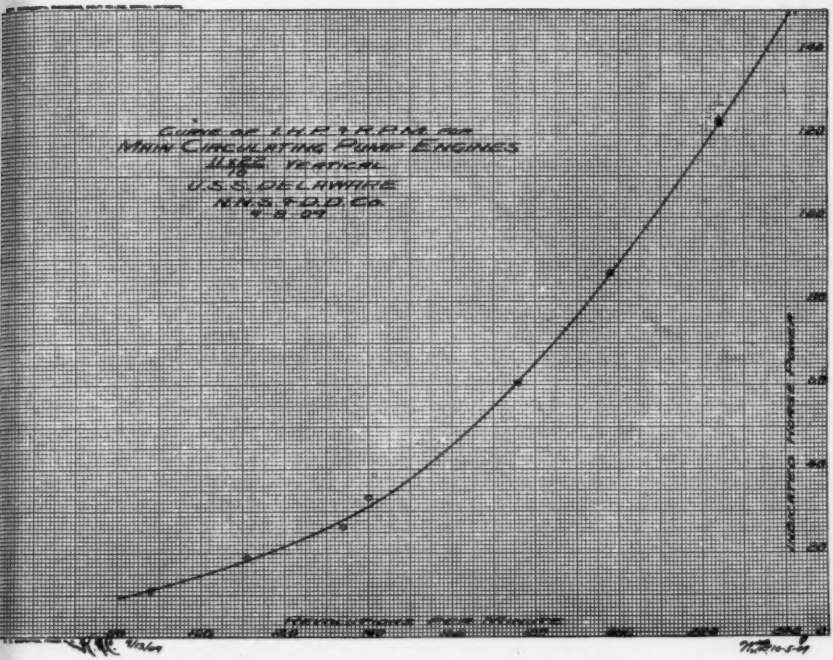
INDICATOR CARDS. SET NO. 4.

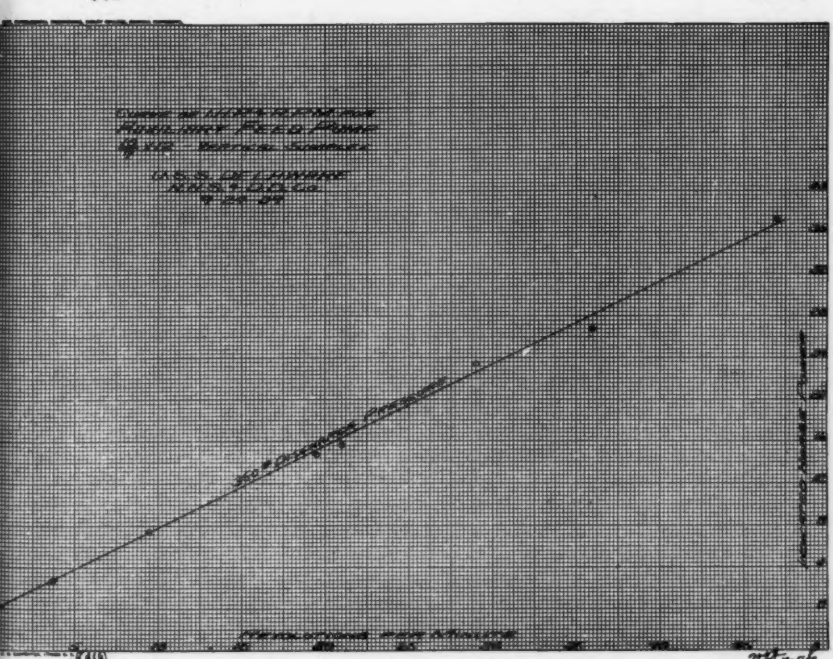

300
280
260
240
220
200
180
160
140
120
100
80
60
40
20
0
M.E.P.
118.5
M.E.P.
54.5
M.E.P.
23.82
U.S.S. DELAWARE.
COMBINED DIAGRAM.
STARBOARD ENGINE.
CARDS, SET No. 4.
M.E.P REFERRED TO L.P CYLINDER = 54.07.

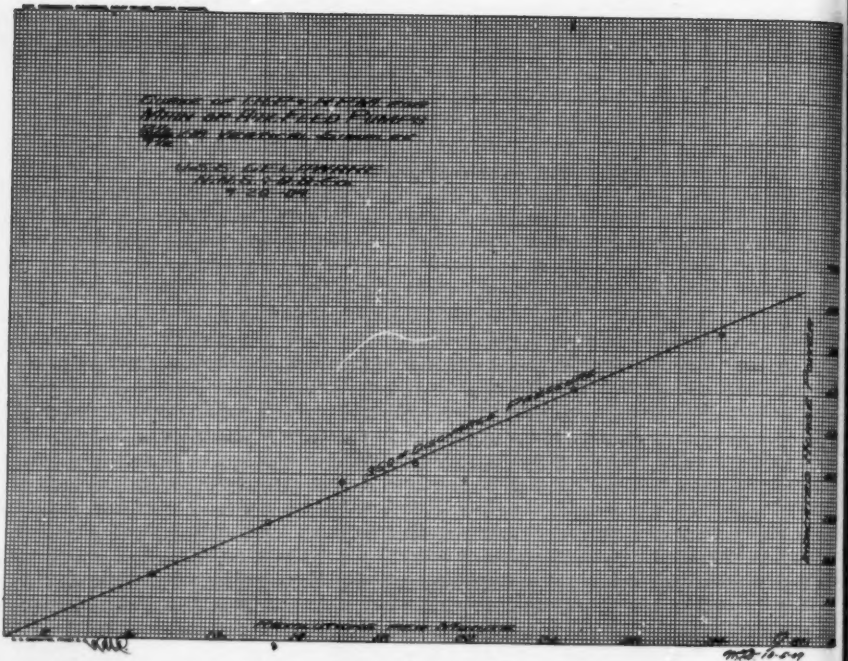

DATA.
SCREW PROPELLERS.
Diameter ..... 18ft 3"
Pitch as set ..... 19ft 9"
Projected area 85.4
Helicoidal area 105.84
Helic. area / Disk area ..... .405
Remarks:
High Power Runs indicated thus. ⊙
Revolutions and Speed Corrected for tide
Propulsive Coefficient
with
without
Appendages.
PROPULSIVE COEFFICIENT.
70
60
50
50
60
70
80
90
REVOLUTIONS PER MIN

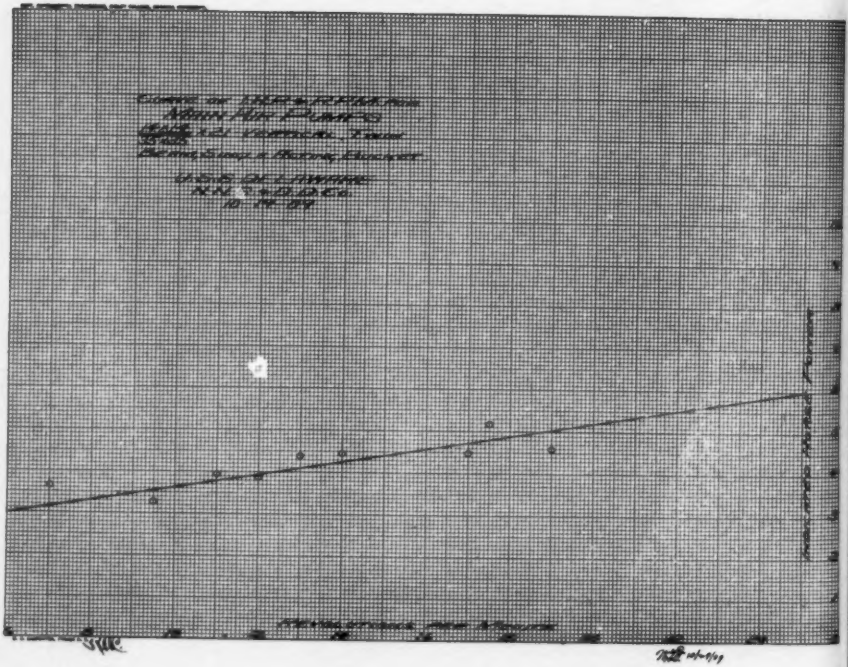

Revolutions and Indicated Horse Power.
Effective Horse Power with Appendages.
Effective Horse Power without Appendages.
INDICATED HORSE POWER.
SPEED IN KNOTS PER HOUR.
27000
25000
23000
21000
19000
17000
15000
13000
11000
9000
7000
5000
3000
1000
22
21
20
19
18
17
16
15
14
13
12
11
10
9
90
100
110
120
130
ER MINUTE.
U.S.S. DELAWARE.
STANDARDIZATION RUNS.
Over the measured mile course
Rockland, Me. October 22nd 1909
Letter No.

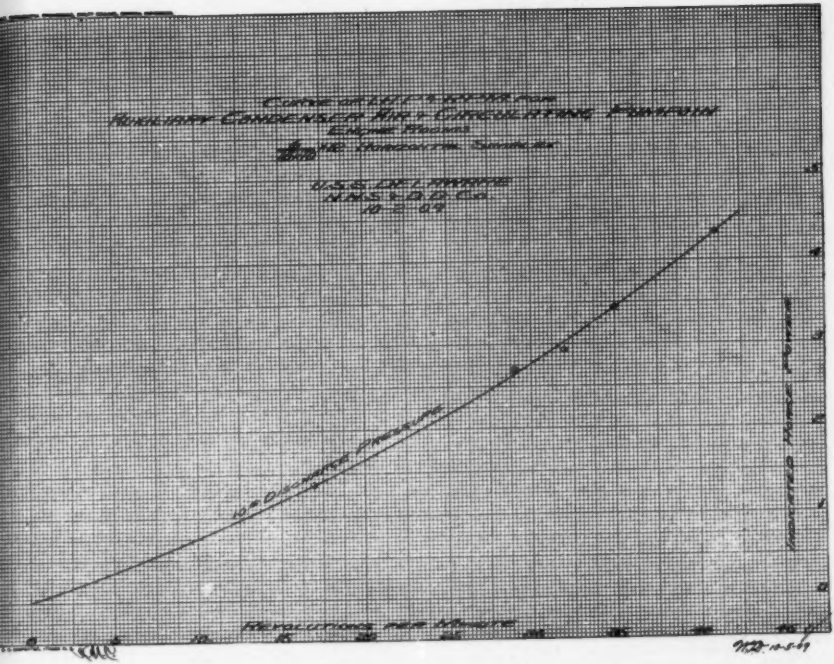


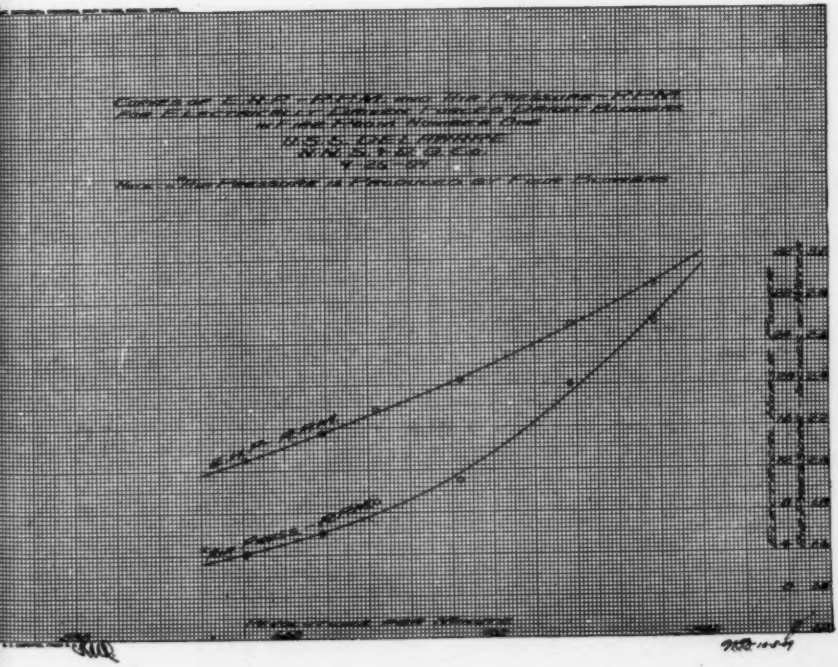

CURVE OF INDICATED HORSE POWER
MAIN CIRCULATING PUMP ENGINES
U.S.S. DELAWARE
N.N.S.B.D.D. CO.
9-8-09
INDICATED HORSE POWER
REVOLUTIONS PER MINUTE

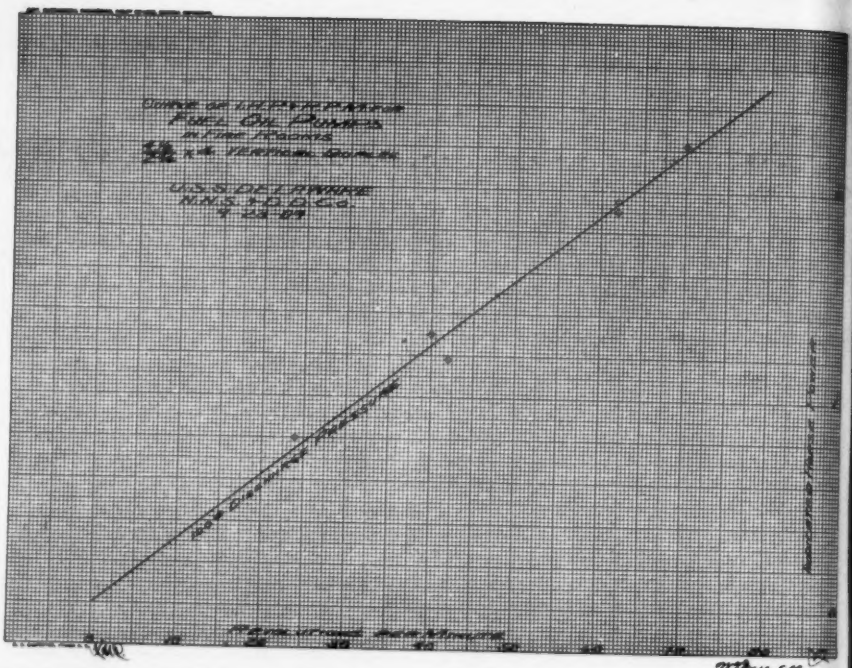

U.S.S. DELAWARE
N.N.S.B.D.D. CO.
REVOLUTIONS PER MINUTE

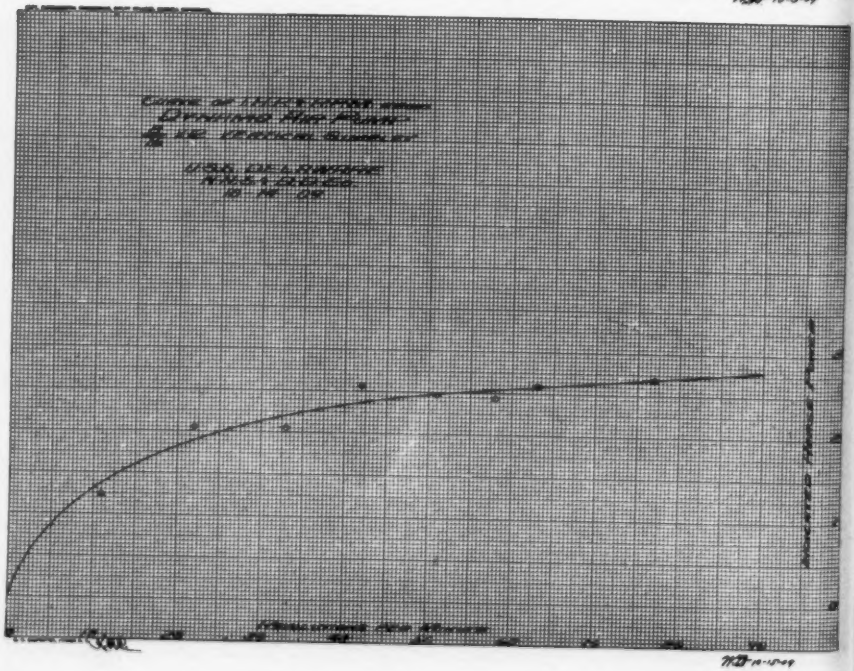


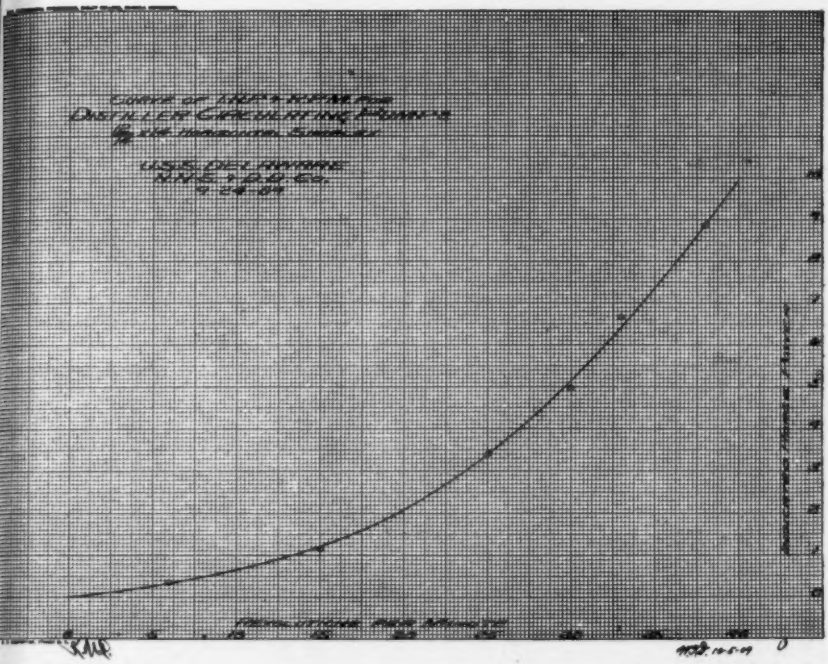


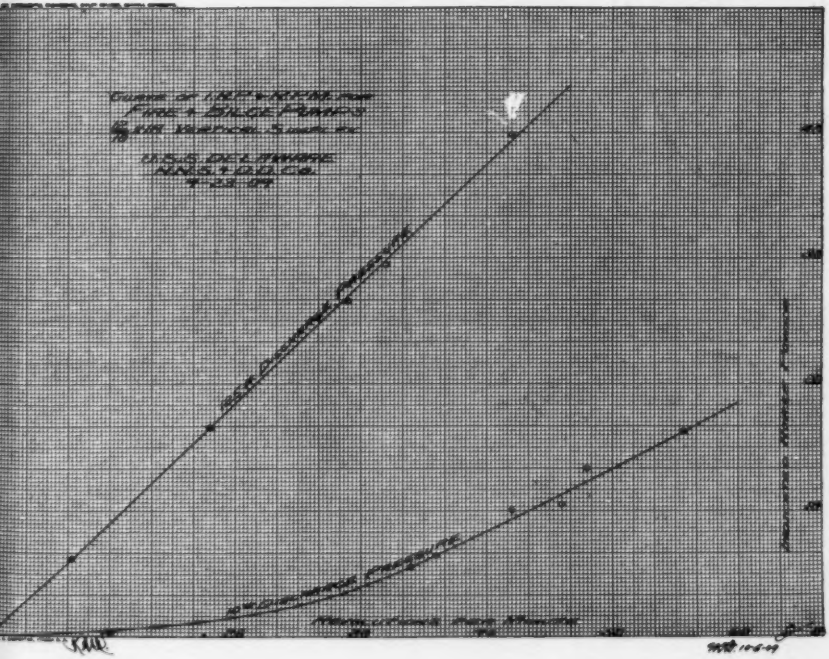


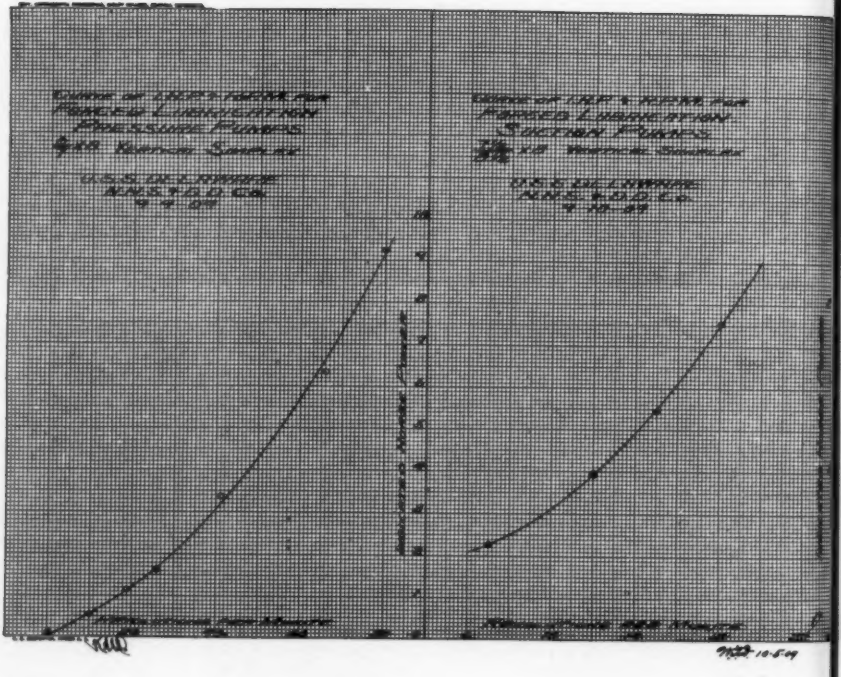



Curve of Performance
Fuel Oil Pumps
at Test Stand
2 x 4 Vertical Duplex
U.S.S. DELAWARE
N.N.S. & D.D. Co.
9-23-09
Brake Horse Power
Pounds Discharged per Minute


Curve of Performance
Drinking Hot Pump
2 x 10 Vertical Duplex
U.S.S. DELAWARE
N.N.S. & D.D. Co.
10-14-09
Indicated Horse Power
Pounds Discharged per Minute


Curve of Heat Input per
Distilled Circulating Pounds
of the Indicated Steam
U.S.S. Del Monte
Nov. 1, 1909
9-24-09

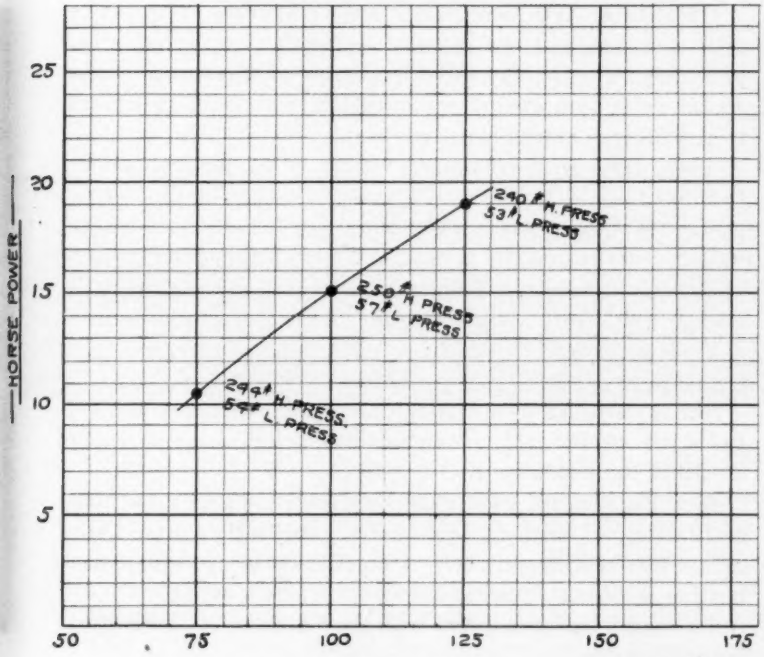

Curve of Heat Input per
Pound of Dried Fuel
of the Natural Draft
U.S.S. Del Monte
Nov. 1, 1909
9-23-09


HORSE POWER
25
20
15
10
5
50

HORSE POWER CURVE

OF STEAM CYLINDER AT ATMOSPHERIC EXHAUST

2 TON ALLEN DENSE AIR ICE MACHINE



H. B ROELKER
41 MAIDEN LANE
NEW YORK QTY

85

THE EFFECT OF SUPERHEAT ON THE ECONOMY OF A MARINE ENGINE AS DETERMINED BY TESTS ON THE STEAM YACHT *IDALIA*.

By Lieut. John Halligan, Jr., U. S. N., Member.

On October 11th, 12th, 13th and 14th, 1909, comparative tests were made of the steam consumption of the main engine, feed, circulating and air pumps of the steam yacht *Idalia*, using saturated steam and steam with from 57 degrees to 105 degrees Fahrenheit superheat.

These tests were conducted by Dr. D. S. Jacobus, under the direction of Mr. W. D. Hoxie, the owner of the *Idalia*, through whose courtesy tests 3, 4 and 5 were witnessed by the writer, representing the Bureau of Steam Engineering.

The *Idalia* is of the following general dimensions: Length over all, 178 feet; length on waterline, 140 feet; beam, 20 feet; depth, 10 feet; draught, 9 feet; gross tonnage, 201; net tonnage, 111.

She is propelled by a four-cylinder triple-expansion engine, $11\frac{1}{2} \times 19 \times 22\frac{11}{16} \times 22\frac{11}{16} \times 18$ inches stroke, the valves of which are all of the piston type.

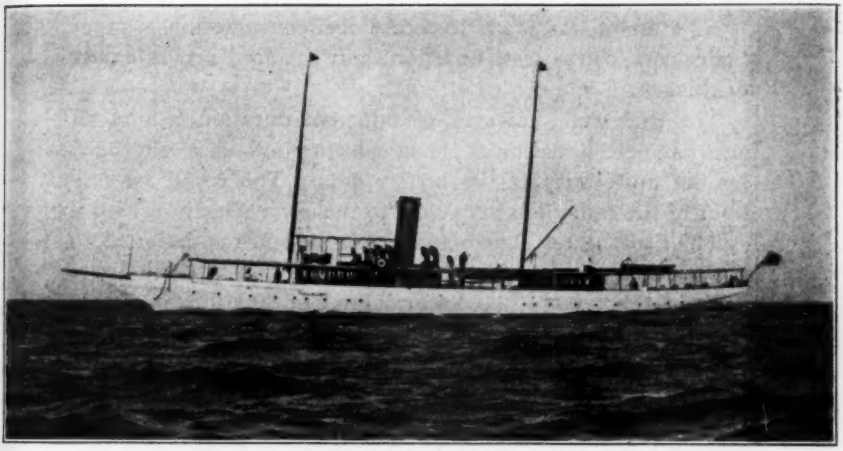
Steam is supplied by a Babcock & Wilcox boiler, burning anthracite coal, with 65 square feet grate surface and 2,500 square feet heating surface. A superheater of 340 square feet heating surface is fitted, as shown in figures 2 and 3.

The auxiliaries connected with the tests are of the following dimensions:

Feed pump, inches, . . .	$6 \times 4 \times 6$.
Air pump (independent), inches, .	$6 \times 12 \times 8$.
Circulating pump (centrifugal), inches,	$5\frac{7}{16} \times 5$.

Under ordinary conditions of cruising about 100 degrees of superheat is carried. Except in the case of the dynamo engine all of the auxiliaries take superheated steam at full boiler pressure. The only lubricant used in the main and

auxiliary steam cylinders is fine graphite mixed with water. On the main engine this mixture is introduced by means of a hand-lubricating pump fitted at the throttle, which supplies all the lubricant required for the four main pistons and their respective valves. No difficulty has been experienced in the upkeep of the plant attributable to the use of superheated



STEAM YACHT "IDALIA."

steam. That the cylinders are in excellent condition is evidenced by the efficiency of the engine as shown on the tests.

The tests consisted of runs under conditions that were practically identical as regards their effect on the results. The main engines and auxiliaries were run at nearly constant speeds. The same indicators, thermometers and gauges were used in all the tests, so that the results are strictly comparable.

The steam used by the main engine, the feed, the air and the circulating pumps (all of which are independent) was determined by weighing the condensed steam discharged from the surface condenser, it being pumped from the hotwell into a weighing tank mounted on platform scales, from which it was discharged into a feed tank. Temperatures and pressures were recorded at intervals of fifteen minutes. Frequent sets of indicator cards were taken, the results of all of which are

shown in the tables. New indicators with outside springs were used. The gross and tare weights of each tank full of water were measured and the net weight of the partially filled tank at the end of each hour was recorded.

The dynamo engine was not run during the tests. The only other auxiliaries—the blower engines—exhausted into the atmosphere.

No attempt was made to record coal consumption or other boiler data, as the tests were too short to afford accurate data of this sort.

The tests were, however, of sufficient duration to give an accurate determination of steam consumption, as is evidenced by the uniformity of the hourly data. The result for any hour of the tests plots very close to the curve shown in Fig. 5.

The superheater shown in Figs. 2 and 3 was designed to provide 100 degrees of superheat; this established the upper limit of the tests. The lower degrees of superheat were obtained by mixing superheated with saturated steam, and by partly shielding the superheater from the hot gases by means of asbestos mill board. It was impossible with the means at hand to secure a lower degree of superheat than 57 degrees.

In the test with saturated steam the superheater was completely cut off from the steam line and vented to the atmosphere. The saturated steam was shown to be dry by a throttling calorimeter.

The results obtained are shown in the following tables and, in the form of a curve, in Fig. 5. It will be noted that the water consumption of the main engine and auxiliaries was 18.3 pounds per I.H.P. per hour with saturated steam, and 15.5 pounds when steam superheated 105 degrees was used.

Expressing these results as heat consumption per I.H.P. per minute we have, respectively, 365.7 and 326.9 B.T.U's. That is, the heat consumption with 105 degrees superheat is 89.4 per cent. of that when using saturated steam. In other words, the tests indicate that there is a net gain of 10 per cent. obtained by using 100 degrees Fahrenheit of superheat. This may be expressed as a saving in coal of 1 per cent. for each 10 degrees of superheat.

SUMMARY OF INDICATOR CARDS.

SATURATED STEAM.

Date. 1909.	Time.	Card.	Mean effective pressure. H.P.	I.P.	F.L.P.	A.L.P.	Water per hour.
Oct. 11.	2:15						
	2:30	1	80	38.0	13.0	12.0	
	3:00	2	75	37.0	12.5	11.5	
	3:15						9,177
	3:30	3	75	37.5	12.0	11.5	
	4:00	4	75	38.0	13.0	12.5	
	4:15						9,214
	4:30	5	86	40.0	13.75	13.0	
	4:45						5,101
	Averages.....		78.2	38.1	12.85	12.1	9,397

Collective I.H.P., 512.34.

SUPERHEATED STEAM, 67 DEGREES.

Date. 1909.	Time.	Card.	Mean effective pressure. H.P.	I.P.	F.L.P.	A.L.P.	Water per hour.
Oct. 14.	2:30						
	2:35	1	78.5	37.0	13.25	12.25	
	2:55	2	84.0	37.5	13.25	12.25	
	3:15	3	81.0	37.5	13.00	12.25	
	3:30						8,571
	3:35	4	70.5	33.0	11.50	11.00	
	3:55	5	80.5	37.0	13.25	12.25	
	4:15	6	75.0	37.0	11.75	11.85	
	4:30						8,290
	Averages		78.25	36.5	12.67	11.98	8,430

Collective I.H.P., 495.2.

SUPERHEATED STEAM, 88 DEGREES.

Date. 1909.	Time.	Card.	Mean effective pressure. H.P.	I.P.	F.L.P.	A.L.P.	Water per hour.
Oct. 14.	9:45						
	9:55	1	89.0	35.0	12.00	11.50	
	10:15	2	85.0	33.0	11.66	10.125	
	10:35	3	91.0	37.0	12.50	11.50	
	10:45						7,965
	10:51	4	95.0	37.0	12.50	12.00	
	11:15	5	95.0	37.0	13.65	12.60	
	11:35	6	87.5	37.0	12.10	12.00	
	11:45						8,291
	11:55	7	85.0	36.0	13.75	12.50	
	12:15	8	85.0	35.0	12.50	12.25	
	12:35	9	90.0	36.0	13.00	12.50	
	12:45						8,447
	Averages.....		89.17	36.0	12.63	11.89	8,234

Collective I.H.P., 521.14.

SUPERHEATED STEAM, 96 DEGREES.

Date. 1909.	Time.	Card.	Mean effective pressure. H.P.	I.P.	F.L.P.	A.L.P.	Water per hour.
Oct. 12.	3:00						
	3:15	1	84.75	...	10.25	10.00	
	3:45	2	91.50	35.00	14.50	12.00	
	4:00						7,842
	4:15	3	93.00	33.75	...	11.00	
	4:50	4	87.50	35.00	13.50	11.50	
	5:00						7,994
	5:04	5	87.00	34.50	13.00	11.00	
	5:26	6	87.70	35.00	12.75	10.50	
	5:30						3,920
	Averages.....		88.57	34.65	12.8	11.0	7,902

Collective I.H.P., 498.27.

SUPERHEATED STEAM, 105 DEGREES.

Date. 1909.	Time.	Card.	Mean effective pressure. H.P.	I.P.	F.L.P.	A.L.P.	Water per hour.
Oct. 13.	2:35						
	3:00	1	...	33.0	11.25	10.25	
	3:15	2	91.0	35.0	13.00	11.50	
	3:35						7,793
	3:38	3	95.0	35.0	12.15	10.75	
	3:54	4	92.5	34.0	12.50	10.80	
	4:10	5	95.0	34.5	12.10	10.50	
	4:29	6	92.5	35.0	12.30	10.90	
	4:35						7,863
	4:48	7	91.0	34.0	12.65	11.00	
	5:05	8	91.0	36.0	12.15	11.00	
	5:24	9	83.5	35.0	11.50	10.80	
	5:35						7,713
	Averages.....		91.4	34,6	12.17	10.83	7,790

Collective I.H.P., 502.15.

SUMMARY OF TESTS.

Date.	Conditions.	R.p.m.	I.H.P.	Water per hour.	Water per I.H.P.	Pressures. Throttle.	1st rec.	2d rec.	Vacuum.	Temp. Feed.	Hotwell.	R.p.m. Air pump.	Circ. pump.
Oct. 11....	Saturated.....	194.3	512.3	9,397	18.3	190	68.4	9.7	25.5	201	116.0	57	196
14....	Superheat, 57°...	191.5	495.2	8,430	17.0	196	66.0	9.2	25.9	206	109.5	56	198
14....	Superheat, 88°...	195.1	521.1	8,234	15.8	201	64.3	8.7	25 9	205	115.0	53	196
12....	Superheat, 96°...	191.5	498.3	7,902	15.8	198	61.9	7.8	25.4	202	111.5	54	198
13....	Superheat, 105°...	193.1	502.2	7,790	15.5	203	63.0	8.4	25.2	200	111.0	45	197

Norm.—These tests and their results are of special interest and value in view of the fact that they

NOTE.—These tests and their results are of special interest and value in view of the fact that they were made on installation in service, and with such refinement and accuracy that the results may be taken as conclusive and as showing the actual economy to be obtained by the use of superheat with marine engines. The tests are probably the first made on an actual installation for the purpose in which such extensive refinements and complete apparatus has been used.—H. C. D.

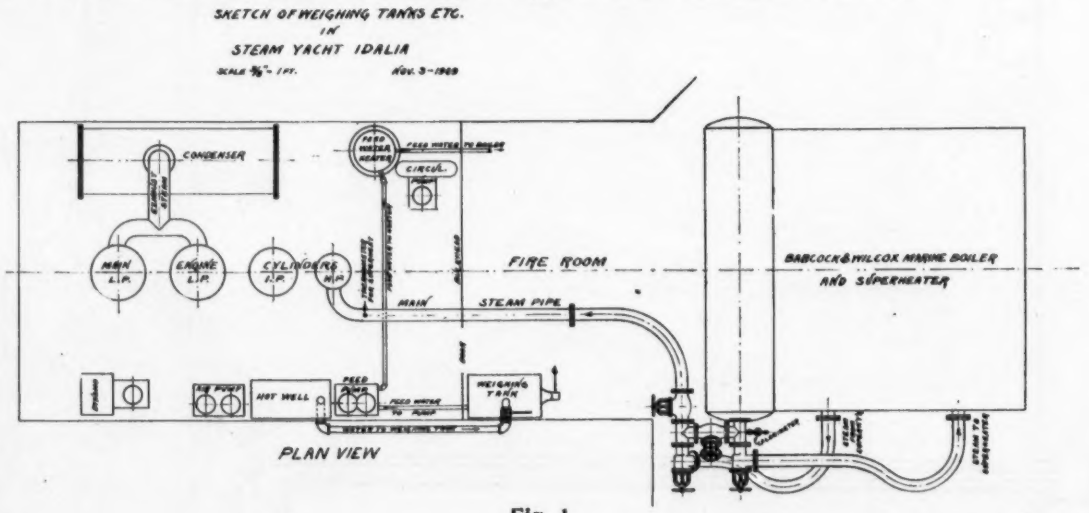


Fig. 1.

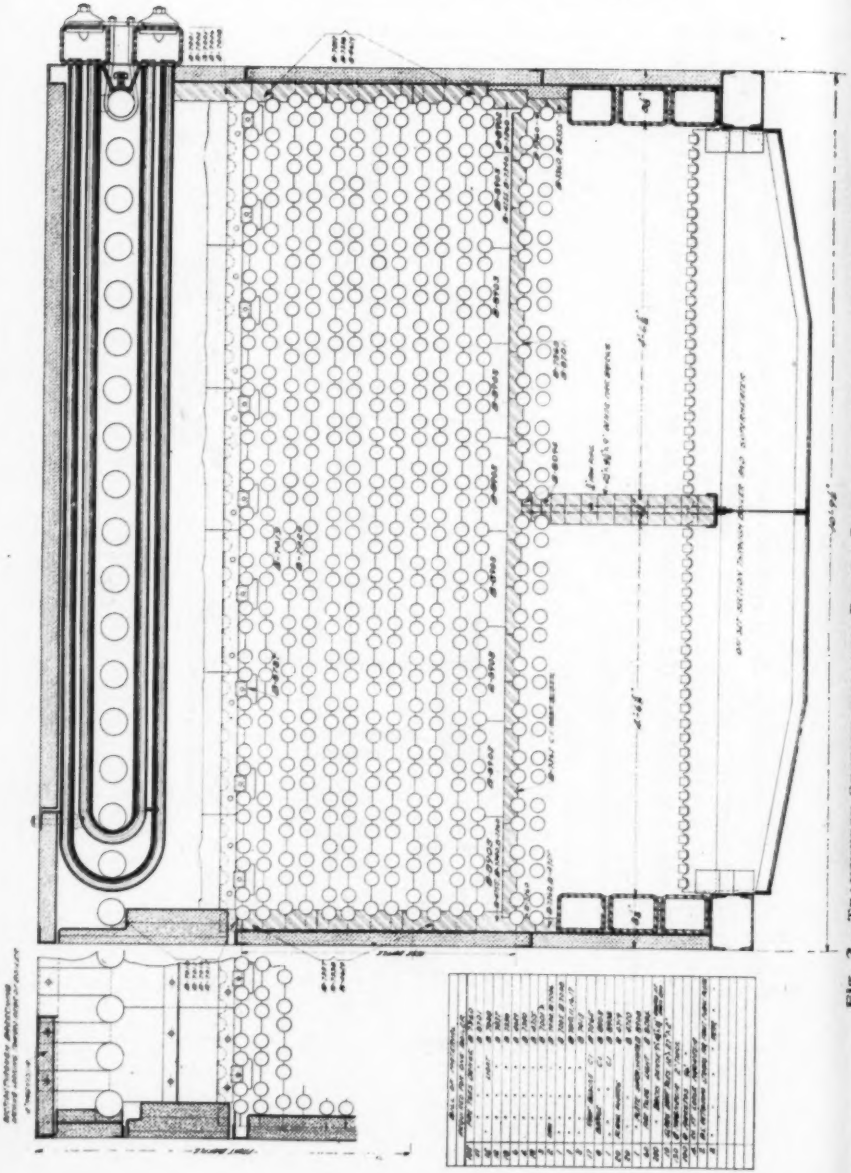


FIG. 2.—TRANSVERSE SECTION THROUGH BOILER SHOWING LOCATION OF SUPERHEATER.

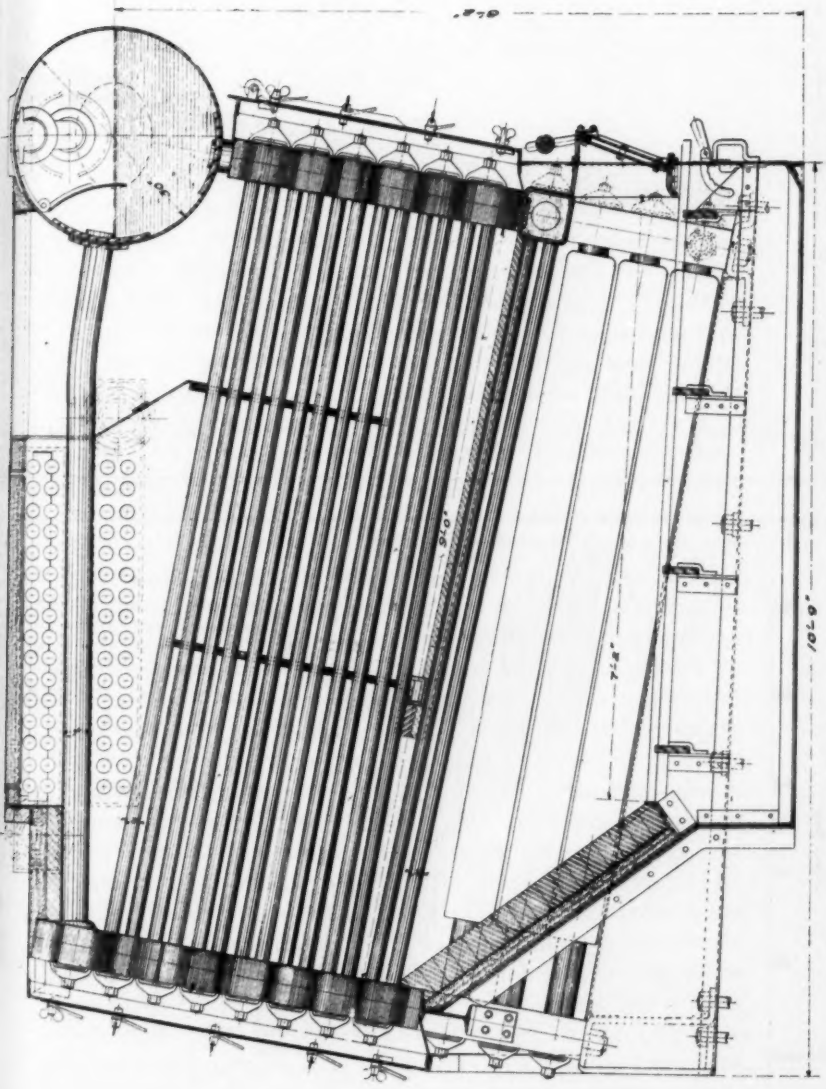


Fig. 3.—Longitudinal Section through Boiler Showing Location of Superheater.

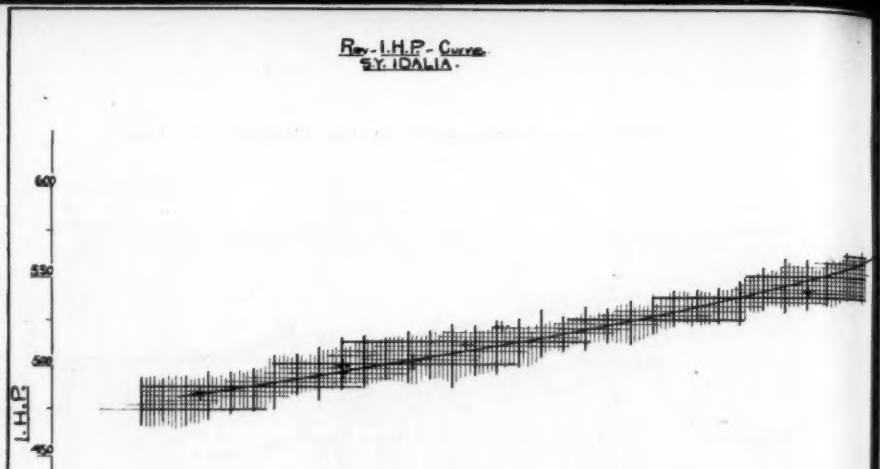

Rev.-I.H.P.-Curve.
S.Y. IDALIA.
I.H.P.
600
550
500
450
400
185
190
195
200
R.P.M.

Fig. 4.

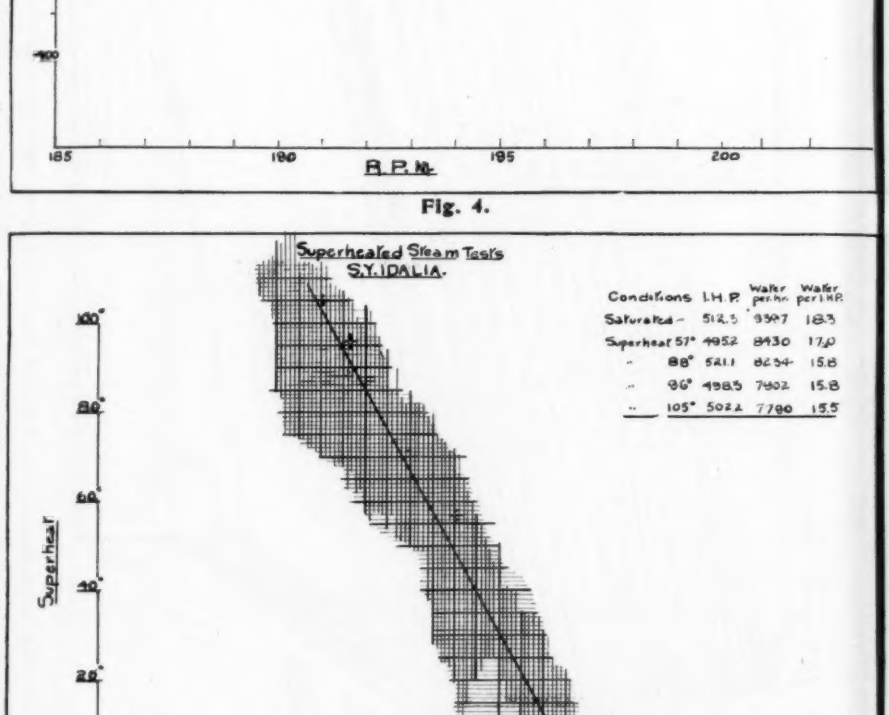

Superheated Steam Tests
S.Y. IDALIA.
Conditions I.H.P. Water per hr. Water per I.H.P.
Saturated - 512.5 9397 18.3
Superheat 57° 495.2 8430 17.0
" 88° 521.1 8234 15.8
" 96° 498.5 7902 15.8
" 105° 502.2 7790 15.5
Superheat
100°
80°
60°
40°
20°
Sat. Steam 0
14
15
16
17
18
19
Pounds of water per hr. per I.H.P.

Fig. 5.

NOTES.

THE PRIMARY ARMAMENT OF BATTLESHIPS.

For the last fifteen years the 12-inch gun has been the standard heavy weapon mounted on all British battleships, and between 1895 and 1905 four such guns formed the primary armament, the secondary armament comprising twelve 6-inch guns; but with the general adoption of the *Dreadnought* type of battleship the number of heavy pieces has been greatly increased, and the individual power and rate of firing has been enormously augmented. In the *Majestics* the 12-inch guns were only 35 calibers long—in the *St. Vincents* 50-caliber weapons are to be adopted. The weight of projectile remains substantially the same—850 pounds—but the muzzle velocity has gone up from 2,300 feet to practically 3,000 feet per second, and the muzzle energy from 33,000 foot-tons to practically 50,000, an increase of over 50 per cent. The size of ships has risen nearly in proportion, while, owing to experience gained in the Russo-Japanese war, systems of protection have been more than correspondingly improved. The question of number, caliber and position of heavy guns for future designs is consequently engrossing a very large amount of attention at the present time, not only in this country but also abroad, where it is felt that the gun has slightly receded from its position relatively to the structures it is required to attack, nor, perhaps, in the case of the pre-*Dreadnought* types, but undoubtedly as regards the later classes of ship. The point at issue is really whether a greater number of guns of the same caliber should be carried, whether by a modified disposition they should all be arranged to fire on either beam, or if a larger caliber is to be introduced. That Admiralties will adhere for ten or twelve years to ten 12-inch guns and a score of 4-inch as a standard armament, much as four 12-inch and twelve 6-inch were ad-

hered to in the *Majestic, Glory, London* and *Exmouth* classes, is extremely unlikely, and the importance of the possible change should not be overlooked. If an increased number of turrets is required it is not hard to foresee the probable arrangement. Little scope for variation exists within the limits of 20,000 to 22,000 tons, owing to the size of the gun mountings and the restrictions of space; four center-line turrets and two broadside turrets arranged as in the *Minas Geraes* are inevitable. It might conceivably be possible to place two center-line turrets sufficiently far aft and forward to admit of two pairs of broadside turrets at each end, but such an arrangement greatly restricts the arcs of fire. If only four turrets are adopted the most suitable arrangement is obviously to place them all on the center line, with the two center turrets arranged to fire over the two end guns. Such an arrangement is, we are aware, different from that adopted in the *Inflexibles,* but these ships are, for all practical purposes, only six-gun ships, the arc of training off the broadside guns on the other beam being very small. If all ten turrets of the *Dreadnought* were to be arranged to fire on one beam a considerable addition would have to be made to the length of ship. This has been accomplished in the *Delaware,* but only at the sacrifice of end-on fire, while the ship has had to be made much longer than the *Dreadnought.*

By no means novel, yet of more than ordinary importance at this juncture, is the suggestion to adopt triple mountings in one turret. Such an arrangement may effect a slight saving in proportionate weight of armor, and to some extent overcomes the difficulty of carrying two more twin mountings, but it has very serious objections. The principal of these is the effect of the recoil on the training of the turrets. In the ordinary twin 12-inch mountings the guns are each some 4 feet off the center line, and the "throw-off" due to the discharge of one gun is some four or five degrees, which has to be made up before the other gun can be fired on the same bearing. With triple mountings either of the two outside guns will be at least 7 feet off the turret axis, and this effect will be

greatly exaggerated. Whatever may be the advantages of the reduced number of turrets to the naval architect, even although they be of greater diameter than the existing turrets, triple mountings are, at present, sure of meeting with the unqualified condemnation of practical gunnery men.

The other alternative is to increase the caliber of the primary guns, which would be a step of far-reaching importance. For one thing, the weight of the armament and ammunition would be enormously increased; for another, the life of the gun would be reduced. Gunnery, however, has never stood still, and of the three alternatives with which naval designers are now faced increase of caliber is most favored by the majority. Even if the number of primary guns is reduced from ten to eight, this would involve an increase of no less than 35 per cent. in the weight of the main armament, and with this and the larger turret diameters involved the size of ships is bound to increase tremendously.

American and German artillerists are credited with ideas of 14-inch and 35-cm. guns respectively, but 13½-inch is the caliber suggested in this country, which, for the first time for many years, would then possess a smaller principal gun than those in use in other navies. A weapon of this nature 45 calibers in length would weigh about 85 tons, compared with existing 58-ton guns; the energy developed at the muzzle would be approximately 70,000 foot-tons. Many objections can be raised to the gun itself, but as a component feature of a first-class battleship it must be regarded from a more general point of view. The size and cost of warships have increased out of all proportion to the relative capacity of their destructive powers, because, although more primary guns are carried, far more has been devoted to features of internal protection, to interchangeability, to greater subdivision and to enhanced weight of machinery. Far more is also done now-a-days in the way of improved means of ventilation, of automatic signals and systems of communication of all kinds throughout the ship, which the growth in armament has gone far to obscure, and all these items add to the fighting power by increasing the

general internal efficiency of the ship, but do not affect the destructive power of the main armament in any way.*

The point of view that it is essential to take when considering the offensive and defensive powers of British ships is that of the conflicting items, of their offensive powers contrasted with opposing armor, and of possible hostile gun fire when comparing their capabilities of defence. Viewed in the light of a *Bellerophon versus* a *Temeraire,* we cannot avoid the feeling that under conditions of actual warfare the 11-inch armor would defeat the 12-inch projectile. Much more so will it defeat the 28-cm. shot, but the 30-cm. belt of the *Nassau* should be still more impervious to the 12-inch gun, and the number of pieces, unless more can be brought to bear at one time, does not affect penetration. The striking energy of a shot is practically in proportion to the cube of the diameter, and armor that might resist a blow from a 12-inch projectile would be unable to cope with percussion 40 per cent. greater. The resisting qualities of armor have undergone little or no increase in the last three or four years, and progress in this direction is liable to be slow, as it appears that manufacturers are closely approaching the limits obtainable with a given thickness. An increase of an inch in belt or turret armor presents little difficulty, as the ship dimensions can be readily increased by the small amount necessary in all directions to carry it, but such a step means that the existing primary armament is even less capable of penetrating the external protection. Moreover, all modern vessels possess very considerable internal protection as well. Undoubtedly the 13½-inch gun would would restore the supremacy of the primary armament, but this cannot be done, except at great expense, even if eight-gun ships, with four center-line turrets disposed, as in the *Michigan,* are adopted. The size of ship must be increased, and consequently the weight of armor goes up; the weight of main turrets and ammunition will be increased 75 per cent. for the same number of guns. This growth of individual cost has

*This seems a bit misleading, since the improved communications, fire-control systems and auxiliary devices very materially add to the destructive power of the main armament, else they would not be fitted.

been accepted before—as, for instance, when the *Venerables* succeeded the *Glory* class, or the *Commonwealth* followed the *Exmouths,* and, in turn, cost 15 per cent. less than the *Agamemnon* and 30 per cent. less than the *Dreadnoughts.* The cost per ton has risen from £70 in the *Formidables* to over £90 in recent ships, and there appears to be every likelihood of a fresh increase. The money will have to be forthcoming if battleships built for the Royal Navy are to be capable of effectually destroying possible opponents, but where from is the business of the Treasury and not of the Board of Admiralty.—"The Engineer."

THE ARMAMENT OF MODERN BATTLESHIPS.

The discussion on the question as to whether the primary armament of battleships should include only one caliber of guns, as in the case of *Dreadnoughts,* or a combination of, say 12-inch and 9.2-inch, or some other lighter breech loader, continues, although with less persistency. But it is becoming of less moment since all Powers have followed our Admiralty in adopting the principle. The views, nevertheless, of foreign writers is of interest. French naval authorities have gone very far in their criticisms on the subject, and "Le Siècle"—the journal of M. de Lanessan, a former Minister of Marine—gives preference to ships having the armament of the *Patrie* and *République* class of battleships, which correspond generally to our *Formidable* class, the predecessor of the *King Edward* class, which itself preceded the *Lord Nelson* and the *Dreadnought.* In Austria experts have written with greater length than show of conviction on the "heresy of the *Dreadnought.*" There is also an abundance of criticism of German origin.

The reasoning which serves mainly as a basis for those writers who are adverse to the *Dreadnought,* especially in the matter of her armament, is that this battleship, given the date of its plans, does not embody the lessons taught by the late

war in the Far East, and takes no account especially of the teachings afforded by the Tsushima engagement. This war, they say, showed very clearly the value of medium-caliber ordnance; this class of ordnance should not have been set aside; it should be reinstated. But the special conditions applicable in this engagement are too often lost sight of. Notwithstanding, however, such adverse theories, all navies are now building *Dreadnoughts*. There are to be found, it is true, differences in regard to the composition and organization of their armament; nevertheless, all the ships are based on the main principles which we followed in the construction of their prototype.

Seeing that the primary object for which a battleship is built is the fighting of an enemy's warship, it is quite clear that it should be designed with this end in view. Modern battleships are protected, both as regards their vital parts and their armament, by hardened armor, the average thickness of which is not below 9 inches. The attack on such an armor with a gun of a caliber equal to the armor thickness resolves itself simply into a loss of ammunition, and such a gun forms nothing but an encumbrance on board a modern battleship. But here the case of Tsushima is again put forward to show that in order to place a battleship *hors de combat* it is not at all necessary to attack its protective armor. The illustration is correct so far as it goes, but the class of Russian battleships in question, and the state in which they reached Tsushima, encumbered with coal, on an uneven keel, and so low in the water that their side armor was frequently below the water line, should not be lost sight of. These conditions do not apply in the least degree to modern battleships in their normal state, and nobody believes that any nation is going to arm exclusively with the view to defeat a fleet of the composition of that which Admiral Rojdestvensky so ably led to Far Eastern waters.

It has also been stated that success will be better assured by demoralizing the enemy through the depressing effect of a shower of explosive shell, rather than by actual losses in men

and damage to their ships. This perhaps may be true when dealing with an enemy whose morale is weak or weakened, but it is a mere theory at best, and there are no facts available tending to afford confirmation for it. Besides, one can never be certain that such an enemy is the one that will have to be fought. But even then this theory does not speak against the usefulness of large-caliber ordnance; on the contrary, for it is now matter of common knowledge that the Russian crews were constantly in deadly fear of the Japanese heavy high-explosive shells. Those from the smaller caliber guns, even when falling "in showers," were far from causing the terror which the heavy 12-inch shells inspired. It has been well said that a number of smaller shells cannot compensate for one large-sized projectile.

Our Navy has always acted on the principle that the very best argument to use when dealing with an enemy is to sink him, and it is when the Japanese or any other navy has acted on this same principle that the success of an engagement has been the more final. There is, therefore, no alternative but to place battleships in a position to use this argument with all its effectiveness. Moreover, the caliber of guns has been established immutably by the protection afforded to battleships; this protection makes it necessary to have resort to guns exceeding 9 inches in caliber, and there are no facts which can be derived from the late war likely to modify this, but there is much to the contrary.

The second point which demands consideration is as to whether one caliber only should be adopted, instead of mixed batteries. It is not difficult to arrive at a decision on this point if the manifold advantages be considered which the unification of caliber introduce in the manufacture of armament, the training of the crews, the regulating and controlling of firing, the provision of ammunition and supplies. One single caliber simplifies everything in regard to armament, and those means alone which are simple ensure success in warfare. What importance, may well be asked, can the firing of a few rounds more per minute have, if they are fired with powders of a dif-

ferent nature, manufactured at different periods, and if the projectiles they carry do not behave uniformly on their trajectory? The question does not resolve itself into the firing of a greater or a lesser number of rounds during an engagement, but in the number of rounds which tell. This result is very greatly facilitated by uniformity in powder charges and projectiles.

There remains the question of caliber. That of 12 inches was a necessity at the time the *Dreadnought* was built, for various reasons. If a smaller caliber had been decided upon, it meant the risk of having an armament insufficiently powerful to cope with a very probable increase made in the thickness of the armor that would have been opposed to it. There was, on the other hand, no evidence available at the time that a larger caliber than 12 inches was necessary. There may be in the future; but when the time does come for an increase in caliber over 12 inches the contingency will be duly met.

Therefore it is clearly evident that the armament of the *Dreadnought,* consisting of ten 12-inch guns, forms a most logical conception. The ship so armed came in its right place in the methodical evolution of shipbuilding, as determined by the lessons taught by experience, battle practice, and fire-control experiments, which always have ruled in the British shipbuilding industry. It is all very well now to devise very cunning means to place twelve such guns on board—some navies find even eight to be a sufficient number; or to decide, in theory, in favor of 13-inch, or even 14-inch guns instead of 12-inch; or, again, to elect to mount three guns in one turret. The fact, nevertheless, remains that we led the way; the course was opened by the *Dreadnought.*

The same organic, ballistic and tactical reasons which speak in favor of a uniform caliber for the main armament have as much, if not more, weight when considering the secondary armament. It is hardly credible that a navy, having simplified the main armament, should be desirous to agree to complications in regard to the secondary one. In the case of the latter, however, the caliber to be selected cannot be decided upon so

readily as is the case with the main armament, for more varied and numerous factors come into play. The secondary armament has several duties to perform. It has, in the first place, to serve as a defence against torpedo attack; this, in fact, is so far its principal object. It may also come into line against auxiliary ships; these may not always be worth a round from a 12-inch gun, although when there are ten of the latter available it is not necessary to be so sparing of their fire as was the case in the past. It can also eventually be of use in a fight between battleships.

From the point of view of defence against torpedo attack, the 3-inch caliber gun at the time the *Dreadnought* was designed had a series of very great advantages; its lightness made it possible to carry a large number; its ammunition, having a very low weight, could easily be stored on board in large quantities; its rapidity of firing also formed a valuable feature. It was well known that the effect of 3-inch gun fire had occasionallly proved somewhat inadequate to stop destroyers during the late war, but the class of projectiles then used was sufficient to account for this, and it was thought that a high explosive shell of better make would improve matters from the point of view of the gun. The arguments, therefore, which spoke in favor of a battery of over twenty 3-inch guns were of a very conclusive order, and they very naturally prevailed.

But matters have progressed under this head also, and destroyers are now built displacing 600, 1,000, and even 1,800 tons. The 3-inch caliber gun now becomes insufficient, and we have adopted the 4-inch; in America they have taken the 5-inch, and the 6-inch is mentioned. It is easy to see that guns of these large calibers, and the two latter especially, can have a greater actual firing efficiency than the 3-inch guns in a fight against battleships, while being also more powerful to repel torpedo attacks. But there is nothing in this which runs counter to the principles which guided us in the building of the *Dreadnought*. It does not in any way mean a return to medium-caliber ordnance in the sense in which that appellation is usually understood, and for the purpose which that class of

ordnance formerly had. The medium-caliber ordnance is far from having produced at Tsushima the effects which are attributed to it. The only respite which the Russian heads of columns had in the course of that engagement was when the Japanese momentarily ceased firing their large-caliber guns, and that can only have occurred when the Japanese armored cruisers, whose armament consisted only of 6-inch and 8-inch guns, with the exception of the 10-inch of the *Kasuga,* tacked ship in order to alter their course. All the accounts of the fight are most convincing on this point, and tend to show that it is idle to demand from medium-caliber guns what they can not give, even sometimes at moderate ranges.

A point which also deserves consideration is this: Formerly medium-caliber guns started from 6-inch upwards. But there appears to be at the present time a tendency to include in this class, or, we should say, among the ordnance which is described as of "medium caliber," both guns which are above 6 inches in caliber and guns which are below that limit. The principal, if not the sole, range of usefulness of this class of ordnance is to repel torpedo craft and destroyers. Rapidity of firing is therefore of utmost importance. In the case of a gun served by hand, the rapidity of firing depends upon the weight of the projectile and the bulk of the charge; and although a 100-pound shell may be considered as a load which can be handled by experienced gunners, the charge of pure nitrocellulose powder required to give to it the necessary muzzle velocity is no doubt more bulky than might be desirable. This probably formed one of the principal considerations when the United States decided to adopt the 5-inch caliber. The 6-inch gun would complicate matters further still in this respect, although it might meet the case using cordite. But in this, as in all matters, there are advantages together with disadvantages, and a selection is difficult.

To sum up, the program which we got out in regard to the armament of the *Dreadnought,* and which we followed, has lost today none of its value. It is being generally followed by the other navies. Stated in abstract, this program established

the following: One single caliber for the main ordnance—that is to say, the minimum which affords a margin of safety as against any possible improvement in armor protection; and one single caliber for the secondary ordnance, the maximum consistent with the carrying out of rapid firing against high-speed torpedo craft.—"Engineering."

THE NUMBER OF SHAFTS FOR TURBINE VESSELS.

A good deal of uncertainty seems to exist among admiralties, naval architects and those responsible for the design of marine turbine installations, in deciding upon the most suitable number of propelling shafts. The conditions frequently govern the choice, as in the case, for instance, of the large Cunarders *Lusitania* and *Mauretania,* where the magnitude of the power to be generated and transmitted at a limited number of revolutions per minute imposed on the designers the use of the highest reasonable number of units. But in the great majority of cases the number of shafts is not subject to such limitation, and it may be interesting to investigate the governing factors which decide the number. It is necessary to state first the general considerations, and afterwards to consider their application in the case of the special types of vessel into which the two principal categories of warships and merchant vessels naturally subdivide themselves. The governing factors are:

1. Turbine efficiency, relative to: (*a*) Weight; (*b*) First cost; (*c*) Space occupied; (*d*) Simplicity of construction and working; (*e*) Subdivision into independent steam units; and (*f*) Variations in demand for power.

2. Propeller efficiency.

3. Facilities for convenient subdivision of engine rooms by bulkheads.

4. Maneuvering qualities of the vessel and safety in breakdowns.

5. Convenience of the internal arrangements of the ship.

such as magazines, gun-mountings, or holds, in relation to screw shafts and tunnels.

Taking these points in order, it may be accepted that, in turbine design, the efficiency under any given conditions normally met with is practically independent of type, and that any of the known and tested turbines now on the market may be so designed as to attain a high degree of commercial efficiency when the designer is free from considerations of weight, space occupied, prime cost, etc. The various types do, however, vary substantially in the degree to which they lend themselves to the realization of high efficiency in unfavorable working conditions without at the same time paying too dearly for it. Weight depends upon two main factors for all types of turbines under equal conditions of steam pressure and vacuum: Firstly, revolutions, inversely to the square of which weight varies approximately; and, secondly, the proportion of the pressure drop utilized in each unit, the amount of blading or number of stages varying directly with this factor. First cost depends mainly on weight, but partly also on stresses on material and complexity of design, considerations which are here outside our scope. Space occupied is very directly related to weight, since the weight of turbines, even of different types, varies very closely as the cube of the dimensions. As to simplicity of construction and working, it has never been shown that complicated constructions have brought any advance in efficiency over simpler forms, such, for instance, as Parsons blading on a drum type of rotor. The opposite extreme is represented by the multicellular types with but one row of action blades per working wheel. The velocity compounded types, with a limited number of stages, represent an intermediate amount of complication. It is true that in all types fine clearances in one direction or another lead to steam economy at the expense of working safety, but the gain brought by going to the danger limit is too small to afford much temptation. The term "subdivision into independent steam units" is used to denote the turbine or group of turbines in which the whole expansion of the steam takes place. As subdivision involves,

for equal efficiency, a much greater amount of blading or number of stages, as will be noted later, evidently for equal weight and equal number of turbines the greatest efficiency is obtained by using the minimum of independent steam units—*i.e.*, by having turbines arranged as far as possible in series. Variations in demand for power affect efficiency of warship turbine installations chiefly by rendering necessary an exceedingly high maximum efficiency (at full speed) in order that the minimum efficiency (at low speed) may not be prohibitively low. Since the omission of cruising turbines, in many recent installations, it has become necessary to raise the full power efficiency by 10 to 12 per cent., in some cases, in order to bring up the low-speed efficiency correspondingly. In cases of demands for reasonable efficiency at fractions of full power below about one-fifth, the present practice is to save some weight by lowering the full-power efficiency, and utilize this saving to provide either separate cruising turbines of the Parsons type, or additional cruising stages in turbines of the impulse type. It has also been proposed to cut out nozzles at each stage of velocity compounded turbines when running at low speeds; but, although this is no doubt a satisfactory solution where revolutions can be kept constant, as in land turbines, it would seem that for marine turbines such an expedient must be rendered futile by the incorrect ratio of blade to steam speed thus obtained, and we do not know of a successful attempt in this direction, although many details have been patented. A further disadvantage is that all nozzles must be reopened before the turbine can be given full steam again.

As regards the second factor—propeller efficiency—every designer of turbine propellers must be economical in his allowance of blade surface, since every addition to it reacts on the efficiency of the turbine itself. It appears, from the general success on trial of the more recent turbine-driven ships, that a satisfactory solution is being reached as to the problem of minimum blade surface for the maximum of revolutions compatible with over-all efficiency. All we need here concern our-

selves with is the number of screw propellers between which the aggregate blade surface shall be subdivided. On purely theoretical considerations the efficiency of screws working in free water under equal hydraulic head is independent of dimensions; and under these ideal conditions any number of small screws are equal in efficiency to one large screw, provided the aggregate area and thrust be the same, and the laws of similitude be otherwise observed. In the propulsion of ships, however, subdivision of the propelling surface among multiple screws renders it necessary to consider the effects due to changed depth of water above the tips of the blades, partial masking of propellers by portions of the ship's structure, influence of the ship's wake, and interference of one screw with another. The actual conditions are consequently far removed from the ideal case just stated. For practical cases of turbine propulsion the choice is limited to two, three or four screws. The efficiency of either two screws or four screws is evidently likely to be higher than that of three screws, owing to the inevitable partial masking of a middle-line propeller by important structural members of the hull (except in the case of torpedo craft without deadwood). The bossing, or brackets, to carry wing screws are usually of lighter construction. Against this is to be set the fact that a center-line screw receives a more axial afflux of water than wing screws working in non-axial convergent streams. The influence of wake is an obscure subject, on which but little data are in the possession of the public. We understand the results of important experiments are shortly to be published; it is at least known that experiments have been made showing a high propeller efficiency when the screws work in the frictional wake near the hull. It is certain that the effect of this is to reduce the revolutions at which any given turbine and screw must work when producing a given maximum thrust, so that the decrease in turbine efficiency is probably greater than the increase in propeller efficiency.

With four screws it is very much easier to avoid the frictional wake, because the smaller diameter and lighter weight

of the screws and shafting render it possible to carry them further from the hull. An incidental advantage is that much vibration can thus be avoided. In an examination of turbine ships it was shown that in all cases those ships with marked propeller vibration had propellers running near the hull, whereas in those where vibration was almost absent the propellers were relatively well clear of the hull. Synchronism is also far less likely to occur with four than with two screws. To these two causes—proximity to the hull, and synchronism—may probably be ascribed the violent propeller vibrations experienced in the case of a recent twin-screw turbine destroyer. Interference of one screw with another may occur in the cases of three and four screws, but not in that of two. It is very difficult to avoid in the case of three screws unless all be placed nearly or quite in the same transverse plane. This has been done with good effect in certain destroyers, and the boats thus fitted are remarkably free from vibration, but such an arrangement is only possible in a limited number of cases. With four screws the wing screws may be placed so far forward, and the inner screws so far aft, that interference is reduced to a minimum. In respect of immersion, three screws are, of course, preferable to two, and four to three. It is thus progessively possible to increase the thrust per unit of surface, owing to the greater hydraulic head; for vessels of limited draught of water, and those subject to severe pitching, such as destroyers, multiple screws offer an important advantage in this respect. It is admitted that the above *pros* and *cons* are not quantitative; still the provisional conclusion seems justified that better efficiency is got from a given screw area divided over four screws than over two screws, and still more efficiency, torpedo craft excepted, than with three screws.

Facilities for convenient subdivision of engine rooms by bulkheads are important factors in determining the number of screw shafts. Subdivision may be carried out either by longitudinal or transverse bulkheads, or by a combination of both. Many shipbuilders prefer only to fit transverse bulk-

heads, for reasons of stability, and a large four-shaft French trans-Atlantic liner now building has the wing screws driven by the turbines in the forward engine room, and the inner screws by the turbines in the after engine room, an arrangement which permits of the vessel navigating in a condition of perfect stability, and maneuvering ahead and astern, even should either of the engine rooms be flooded. With transversely-arranged engine compartments any number of shafts are equally convenient; the German and French destroyers, with Parsons turbines, afford instances of the application of this arrangement to triple screws; the American scout *Chester* and the liner above referred to, to four screws; and the German twin-screw destroyers, with both Curtis and Parsons turbines, to twin screws. Longitudinal bulkheads are resorted to when transversely-arranged engine rooms demand too much space in the length of the vessel. In British four-shaft warships a single middle-line bulkhead has been the rule till recently, a commencement having just been made with two longitudinal bulkheads, an arrangement similar to that fitted in the big Cunarders. Treble screws could similarly have their engines fitted in three parallel engine rooms, though we can recall no instance of this. Hitherto the engines for treble screws have been fitted in single undivided engine rooms, except in the cases where transverse bulkheads are built, as mentioned above. Twin screws, when space for transverse division (and "staggering" of the turbines) is lacking, impose the middle-line bulkhead, which seems an inherent disadvantage, perhaps as great as that of the single engine room used so largely in triple-screw installations. It is very commonly found in recent arrangements that separate condenser and pump rooms are provided; but though this reduces the size of main engine rooms, it does not affect the choice of number of shafts.

Maneuvering qualities of the vessel are, of course, vitally affected by the question we are dealing with, and must be considered. The ideal which the officers of a ship are entitled to enforce upon the consideration of the designer of the ship and machinery is a vessel which will have her rudders beneficially

influenced by the streams of water projected by the screws, and which can at the same time be handled well without steerage way, or if the steering power be temporarily lost. From the steering point of view, the best arrangement for large ships seems to be four screws with two rudders acting in the wake of the inner screws. In small ships three screws and one rudder give exceedingly good results. Unfortunately, with twin-screw vessels the shaft centers are necessarily too far apart to allow of twin rudders in the same line. These ships never steer as well as the three and four-shaft ships above mentioned, having respectively one and two rudders. For handling without steerage way, or for steering without the use of a rudder, it is, of course, necessary that all the screws on each side of the vessel should be able to work independently ahead and astern. In this respect it is evidently better that the number of shafts should be even—*i. e.*, two or four—the triple-screw arrangement offering a considerable handicap. There will evidently be no difference between a two and a four-shaft arrangement.

The question of safety in breakdowns and navigability of the vessel after loss of one or more screw-propellers is favorably influenced by a multiplicity of screws, the loss of one of which diminishes by a small fraction only the available propelling power. A good illustration of this is seen in the performance last year of the *Mauretania,* when she made several excellent passages across the Altantic, carrying mails and passengers, with only three screws working; the fourth being removed owing to a hull defect in its neighborhood. Had this vessel been fitted with only three or two propeller shafts it would, of course, have been necessary to have laid her up for lengthy repairs at the season of the year when the trans-Atlantic passenger traffic was at its height.

The subject of the relation of multiple shafts and convenience of the internal arrangements of the ship is as important almost for merchantmen as for naval ships. In warships it is usually extremely inconvenient for a middle-line shaft to pass underneath the after gun mountings and magazines, and for this reason two and four-shaft arrangements are usually pre-

ferred by the ship designer. There is probably not much to choose between them, because the stern tubes of the outer shafts are so short and so far outboard that they need not be considered in this connection. For cargo vessels shaft tunnels are, of course, a great inconvenience. Twin-screw turbine vessels have their shaft centers raised much higher than in the case of reciprocating-engined ships, owing to the diameter of the turbines necessary for the low revolutions imposed by the system, and these high tunnels afford almost as great, or even greater, obstruction in the after holds as would three lower tunnels. With four shafts the wing tunnels are very short, and the space occupied by them is not very useful for other purposes, while the inner tunnels are much lower than those in a twin-screw turbine vessel, because the four-shaft turbines are of less diameter. In the case of fast Channel steamers and similar vessels, where cabin accommodation is usually fitted in the 'tween decks, no inconvenience is caused by leaving the whole of the after hold open, and using it as a common shaft tunnel for all the shafts. In this case the number of shafts adopted makes little difference.

We have now dealt in detail with the five governing factors enumerated at the commencement of this article, and may now briefly summarize the conclusions we arrive at with respect to the question of the choice of number of shafts for turbine vessels of the various types. The various types of ships coming under consideration may be divided broadly as follows:

Warships.—(*a*) Battleships and large cruisers; (*b*) small cruisers and scouts; (*c*) destroyers; (*d*) torpedo boats.

Merchant Ships.—(*e*) Large fast liners; (*f*) intermediate type of liner; (*g*) Channel steamers; (*h*) tramp steamers.

Considering, first, warships, it may be said that the very high powers now being fitted in battleships and large cruisers govern to a large extent the question of choice of the number of propellers; and it is worth noting that the most recent cruisers designed or ordered rival the big Cunarders in respect of power developed, being of 65,000 to 70,000 horsepower.

When 15,000 horsepower or upwards has to be transmitted through a single shaft at revolutions not exceeding 300 per minute, he would be a bold man who would propose to reduce the number of shafts to three or two, and transmit 20,000 or 30,000 horsepower through each at proportionately reduced revolutions. For cruisers of less power the question may arise; but as twin screws are still excluded by the magnitude of the units involved, the three-shaft alternative alone would be practicable. The disadvantages of the treble-screw arrangement have been noted in the earlier part of this article with reference to propeller efficiency, facilities for convenient subdivision of the engine rooms by bulkheads, maneuvering qualities of the vessel, safety in breakdowns and convenience in disposition of magazines and gun mountings, all of which tell heavily against three shafts.

In the case of modern battleships, where the power is rapidly creeping up to, and even attaining, 30,000 horsepower, the conditions are much the same as in the above-mentioned moderate-sized cruisers, and the increasing use of twin rudders removes the last point remaining in favor of treble screws, viz: the question of the steering qualities. There is no experience of large independent units working on the twin-screw system; but there would be no difficulty in fitting a separate high-pressure and a low-pressure turbine coupled together on each shaft. When this is the case there would be four main turbines in the ship—two high-pressure and two low-pressure; if, instead of coupling them tandemwise, each was arranged to drive an independent propeller shaft, making a four-shaft arrangement, the revolutions could be increased 40 per cent. for the same over-all efficiency, and with a corresponding diminution brought about in the weight and space occupied. This might lead also to a reduction of length of engine room, because the high-pressure turbines may then be fitted abreast of the low-pressure turbines, instead of ahead of them.

In small cruisers and scouts the designer has a freer hand, and, as a matter of fact, turbine vessels of this description have been built with two, three and four shafts. Assuming equal

over-all efficiency of the installation, the two-shaft arrangement must necessarily be considerably heavier than either of the others. The three-shaft arrangement does not lend itself to convenient subdivision of the engine room, and the efficiency of the center screw is likely to be prejudiced, though in some of these vessels the deadwood is cut away almost as much as in destroyers. The difficulty as to gun mountings and magazines disappears, but, on the other hand, turbines of large diameter are inconvenient, because the depth under the protective deck in such vessels is limited, which makes repair work more difficult. For a twin-screw arrangement two systems are in vogue: either the turbine is a complete unit in one cylinder, as is the case in the United States scout *Salem,* and as is proposed, we believe, for H.M.S. *Bristol;* or a high-pressure and a low-pressure turbine are mounted on the same shaft tandemwise, as in the German cruiser *Mainz.* In the former case the designer is strictly limited to the number of stages he can provide for the utilization of the pressure drop of the steam, to obviate the danger of overloading a comparatively thin shaft having a large span between bearings. In the latter case—in the use of four turbines for driving two shafts—the groups become of excessive length, and the complication of the engine room is exactly as great as if four shafts were used. The argument in favor of increasing the revolutions 40 per cent. by coupling each turbine to a separate shaft, and reducing weight and space, becomes very strong. All twin-screw arrangements have the further inherent disadvantage, from the shipbuilders' point of view, of necessitating the fitting of the bulkhead in the middle line if the engine room is to be divided longitudinally, which may involve difficulties in connection with stability. The balance of advantages therefore seems to incline to the four-shaft arrangement.

The use of four shafts enables the revolutions to be so increased that the diameter of the turbines is proportionately reduced. The saving has been indicated as entirely referring to the diameter, but, of course, it could equally be applicable to the length of the turbines, or both diameter and length might be

kept the same, and a considerable increase in turbine efficiency might be secured by higher revolving speed. Irrespective of the method, the profit to be obtained by the use of four shafts instead of two is very great. Supposing the lengths of the turbines to be the same, the two-shaft arrangement requires 40 per cent. greater turbine diameter for the same efficiency, and the weight of the parts affected by this change would be double. The amount of steam and exhaust piping is identical in both arrangements. It therefore follows that: (1) For equal weight the four-shaft arrangement is much more efficient; (2) for equal efficiency the four-shaft arrangement is much lighter; (3) the number of turbines and the amount of steam and exhaust piping are the same in both arrangements; and (4) the only additional complication of the four-shaft arrangement is that of the two extra lines of shafting and two extra propellers. This may be considered to be set off by the additional safety of the vessel should any accident happen to the propeller or machinery, thus enabling her to proceed with three shafts and three-fourths of the power after the accident, instead of one shaft only, or one-half of the power, as in the case of the twin-screw arrangement.

In the class "destroyers" we may include all vessels from the type of H.M.S. *Swift* down to torpedo boats, which latter term may be used to describe boats of 250 tons downwards. For the very largest vessels, of which the *Swift* is at present a unique example, the magnitude of the power involved—something like 50,000 horsepower in her case—again suggests or imposes considerable subdivision, so that the question of twin screws would scarcely arise, the choice being between three and four shafts. Weight, again, plays such an important rôle in such vessels that some arrangement of series working of the turbines seems necessary to secure efficiency without overloading the vessel. The more moderate-sized destroyers, from 1,000 tons down to 500 tons, present great diversity of turbine design; two, three and four shafts all having been adopted, though two shafts have only been fitted so far to boats up to 650 tons. In the latest British boats, however, four shafts

will be fitted. For moderate-sized boats the use of twin screws appears to be justified, although it does not offer any advantages over other systems, and has two special disadvantages, viz: that the large size of the propellers makes it impossible to carry them at a sufficient distance from the hull to avoid vibration, and that the large diameters of the turbines, due to low revolutions and the limited headroom in such vessels, make it impossible to lift the covers and rotors for examination and repairs without recourse to the facilities of a dockyard.

Torpedo boats have, up to the present time, been universally fitted with three shafts, as in the smaller classes of these boats the turbine is approaching the lower limit of its application to marine work. Owing to the smallness of the powers demanded it becomes increasingly important that the turbines should be arranged, as far as possible, in series, in order that the steam may flow through the maximum possible amount of blading. These boats are therefore usually arranged so that the turbines working all three shafts receive the steam one after another. The weight of the installation already compares unfavorably with reciprocating engines, and the efficiency is not much higher, the justification for the adoption of turbines being rather in the smoothness and security of working, and in the absence of wear and tear. Whether such boats could be installed with twin-screw arrangements, and turbines taking independent steam supplies, is somewhat doubtful; but were it possible, they would also suffer from the two disadvantages noted in the case of the twin-screw destroyers, viz: difficulties with propeller vibration and the trouble of overhauling the turbines without recourse to external appliances.

Turning now to merchant ships, we take first large, fast liners, which may be considered from exactly the same point of view as the largest class of cruisers. In fact, their machinery differs very little, except that owing to the heavier displacements, the revolutions are lower and the diameters and weights greater. Subdivision on at least four shafts is thus still more imperative. Such intermediate types of liners as have been fitted with installations of turbines have hitherto been on the

three-shaft system, with the exception of the American turbine steamer *Creole,* which had twin screws, and does not appear to have proved successful. Owing to the very low revolutions of the screws which can be obtained in this class of vessel, it is at present necessary to use more than two shafts if the difficulties of space occupied and the excessive dimensions and weight of turbines in proportion to the power are to be avoided. In spite of the disadvantages which the three-shaft arrangement is exposed to in connection with engine room, bulkheads, shaft tunnels and the low efficiency of the center screw, four-shaft arrangements have not been experimented with for such vessels, though, if the increased number of turbines were not an objection, they could very easily be carried out by the use of high-pressure, intermediate-pressure, and two low-pressure turbines, each driving a separate shaft. Indeed, this arrangement is already being carried out in the case of a large, fast liner building in France, but with the object of increased efficiency rather than that of saving weight or space. The slower, intermediate type of liner falls within the domain of the combination system of reciprocating engines and exhaust-steam turbines, a subject not included within the scope of this article.

In the case of Channel steamers the choice lies between two and three screws, and is not burdened by considerations of great importance in connection with weight, subdivision of engine room, or shaft tunnels. The very fine lines of such craft allow of a fair efficiency from the center screw. Certainly the three-shaft arrangement is preferable to the twin-screw on the score of weight and diameter of turbines, and also in freedom from propeller vibration if the screws be kept well away from the hull, which it is easier to do in the case of treble screws than of twin screws. A commencement is now being made with different systems of gearing for the application of turbine engines to tramp steamers. In such cases the use of twin screws, allowing of 40 per cent. more revolutions of the propellers, would evidently simplify the gearing problem and add to the maneuvering qualities of the vessel; but as it also appears possible to reduce the turbine speed by a

single transmission gear down to that of the normal revolutions of a tramp's single-screw propeller, it is highly probable that the single-screw arrangement will be retained for such vessels.—"Engineering."

NEW BRITISH SCOUTING CRUISERS.

The Admiralty critics have taken exception time and again to what they regard as the unjustifiable delay in the building of light cruisers to checkmate the corresponding vessels of Continental powers. Germany, for instance, has for some time been laying down, with unhesitating regularity, two such vessels per annum, and with these they have, as is well known, been making exhaustive comparative trials of various systems of propelling machinery, a practice which they still continue. They have completed since 1907, or are now building, thirteen modern cruisers, ranging in displacement from 3,200 tons to 4,230 tons, and in designed speed from 23 to 25 knots, while for armament they have twelve 4-inch guns and eight machine-guns, and depend for protection on a protective deck. Such vessels are the eyes of a fleet, and many have been anxious as to whether the relative position in the German and British fleets was being maintained satisfactorily in respect of security, or in the speed with which knowledge of the position of an enemy could be communicated to the officers and commander of the squadron. There was the disquieting fact that, with the exception of the eight scouts, which are not comparable in respect of size or gun power with the German vessels, although of 25 knots speed, we had only four cruising vessels of more than 21 knots. About two years ago, however, the Admiralty recognized that the psychological moment had arrived for the strengthening of our Fleet in this type of ship, and laid down the first of practically a new class in the *Boadicea,* which has just been completed, and exceeded a speed of 25 knots on her trials. Following this came the *Bellona,* a sister ship. Both these vessels are 385 feet in

length, 41 feet in beam, 13½ feet draught, and displace 3,300 tons, and with Yarrow boilers and Parsons turbines develop 18,000 horsepower. It was felt, however, that the armament of only six guns of 4-inch caliber was insufficient for effective reconnaissance work, and last year there were laid down five vessels, which were given the names of cities traditional in the British Navy. The first of these new "City" cruisers was launched September 31, and the others will follow at short intervals during the next two months. This vessel, the first to be launched, is the *Glasgow*, built by the Fairfield Shipbuilding and Engineering Company. The *Gloucester* will be the next to be launched, as she is in an advanced stage at Messrs. William Beardmore & Co.'s Works, Dalmuir. Contemporaneously the *Liverpool* will be floated from the Vickers Company's works at Barrow-in-Furness. The other two vessels are the *Newcastle*, building at Elswick, and the *Bristol*, which is being constructed at Clydebank by Messrs. John Brown & Co. The difference in the date of launch is largely due to the varying practice of the contractors; owing to the minute subdivision of machinery compartments, some of the firms are placing the boilers, or parts of the machinery, on board before the launch, in order that the bulkheads may be riveted and water-tested before the vessels are afloat. The Admiralty are now adjudicating upon the tenders for four more of these "City" cruisers, and Pembroke is making progress with the building of two more *Boadiceas*. Within two years, therefore, we shall have laid down thirteen high-speed cruisers, all of which will exceed 25 knots in speed. This goes some way to make up any deficiency which may exist in this type of cruiser.

Provided the tactical efficiency of the Fleet has not been jeopardized at any time by the absence of these high-speed scouting cruisers—and as to this we have had the assurance of the Committee of Defence of the Cabinet—the Admiralty are to be commended, because delay has brought distinct advantages, particularly in the evolving of a satisfactory system of propelling machinery and armament. The machinery of these

vessels is, undoubtedly, a most important element in the design, as high speed and its maintenance at sea with a high degree of economy, to ensure a wide radius of action, are outstanding desiderata in effective scouting. The efficient use of oil fuel, without the issue of smoke from the funnels, the capacity for sudden increase in power which the water-tube boiler gives, and the facility with which acceleration in speed and the safety with which overloading can be resorted to with the turbines, confer enormous advantages. In regard to weight, too, the turbine machinery has simplified the problem of the naval constructor. In the earlier cruisers with reciprocating engines, the steam consumption was rarely under 18 pounds per indicated horsepower per hour, and then under forced-draft conditions. In the new vessels the guaranteed steam consumption is only 13 pounds, and in the case of the cruiser which is being fitted with the Curtis turbine, using steam with a slight degree of superheat, the guarantee is stated to be 12½ pounds. As a consequence of this there is an appreciable reduction in the boiler power required to maintain full speed. All the "City" cruisers have twelve boilers, with a collective heating surface approximating 50,000 square feet, and a grate area of close upon 900 square feet, a ratio of practically 55 to 1. The machinery for the new ships, it is understood, will only slightly exceed 1,000 tons in weight, so that 22 horsepower is being developed per ton, and that, too, without resorting to a high degree of forced draft. In the scouts the weight of boilers and reciprocating machinery was reduced to a minimum, and the power per ton in earlier high-speed cruisers rarely over 20 indicated horsepower, even when the machinery was severely pressed. The assumption is therefore justified that the power and speed of the new vessels will be attained with a greater degree of reliability . Indeed, the modern type of warship involves only a fraction of the anxiety which weighs upon the engineer responsible for long steaming at high speeds in ships with reciprocating engines, however well they may have been designed.

The *Bristol,* as is now known, is being fitted with Curtis

turbines by Messrs. John Brown & Co., and it is not improbable that this system will also be applied to one or more of the four vessels soon to be ordered. One of the twenty destroyers just ordered is also to be fitted with this installation, which enables twin screws to be fitted. There is just a possibility, too, that in another of the new cruisers the Parsons system will be adapted for twin-screw propulsion. It is likely also that the Parsons new partial-flow reaction turbine will be applied. The other vessels will have the normal Parsons reaction turbine system, the high-pressure end being extended to improve economy at cruising speeds. With the combined impulse and reaction turbines and the normal Parsons system four shafts will be used. The arrangements of engine rooms afford a striking example of the convenience in subdivision, which was referred to at some length in a recent article on the choice of number of propellers (See following). There will be three longitudinal bulkheads in each engine room, the shafts driving the two inside propellers being in the central compartment, and the turbines working the outer propellers in the outer starboard and port compartments. Each shaft will have one ahead turbine, the two on the port side being in series, and the two on the starboard side similarly treated for steam-distribution purposes. There will be two separate astern turbines on the shafts, having the high-pressure ahead turbines, while the low-pressure ahead turbines will, as usual, incorporate the astern turbines. The condensers will be placed abaft in two separate compartments divided by a center-line bulkhead. The Curtis arrangement admits of each shaft having turbines constituting a separate unit, and there will only be two engine rooms divided by a center-line bulkhead, as in all twin-screw cruisers. The engine rooms, as well as the boiler rooms, will have the protection afforded by coal bunkers, as well as by heavy protective decks.

For their size these cruisers are more heavily armed than preceding ships. The earlier vessels were 450 feet in length, 47 feet in beam, and at 15 feet 3 inches displaced 4,800 tons. The ships soon to be ordered will, it is understood, be of the

same length, but of 48 feet beam, and with this slightly increased beam will displace over 5,000 tons. They will have powerful 6-inch bow and stern-chasing guns, and in addition ten 4-inch guns. These are undoubtedly more effective weapons than those placed in the preceding ships, and the number of machine guns will also be increased. Their speed is to be 26 knots, but looking to past experience it is likely that this will be exceeded. The later German vessels to which we have already referred are 388 feet long and of 46 feet beam, displacing about 4,250 tons. Their speed is 25½ knots, but they are only fitted with twelve and fourteen guns corresponding to our 4-inch guns. The new "City" class thus marks a distinct advance in every respect, and notably in radius of action. Their cost, too, cannot be regarded as excessive, as they promise to work out at not much more than £300,000 each, ready for service.—"Engineering."

STEAM TURBINE GOVERNORS.

Steam turbines may be governed in several ways, but the method most commonly employed in this country is that of varying the period of steam admission, and is known as blast governing. This arises from the fact that the majority of turbines in use in England are constructed on the Parsons principle, but on the Continent and in America, where the Curtis, Zoelly, Rateau, De Laval and other types are in prominence, different means of governing are more often met with. In the blast method, as is now well known, steam is admitted to the turbine by a double-beat balanced valve, in puffs of long or short duration, according to the demand for load. When the load is light the valve opens for short periods and remains closed during a greater part of the time, but as the load increases, the period during which the valve is closed becomes shorter, and the turbine consequently receives more steam. The mechanism employed by Mr. Parsons for operating this valve is too well known to need description, but

slight modifications have been introduced by the various firms who have taken up the manufacture of this class of turbine.

So far as the running is concerned blast governing has proved eminently satisfactory, the turbines responding remarkably well to sudden and large fluctuations in load. The speed variation from no load to full load can be kept within very fine limits, and, provided the governor is in order, hunting troubles should be unknown. It is sometimes argued that the relay used for operating the poppet valve involves considerable complication; but this is more imaginary than real. That such a governor is a little more complicated and expensive to construct than one of the ordinary throttle-valve type is, of course, quite obvious; but the introduction of the few additional parts gives the charge engineer but little, if any, more trouble. Usually when a relay-operated governor fails to work satisfactorily the fault can be traced to the relay plunger sticking, and by removing and cleaning it sluggish governing is generally cured. The object of admitting steam to the turbine in puffs instead of in a continuous stream is to prevent throttling at light loads, the idea being that the full benefit of the high-pressure steam can be secured at all loads. Some turbine manufacturers, however, seem to have come to the conclusion that the blast method has very little, if any, advantage over the plain throttling system; and throttle valves actuated by an ordinary fly-ball governor are to be found in use. In the Willans-Parsons steam turbine, for example, a throttle-valve governor is used, and apparently with very satisfactory results. On the other hand, Messrs. Brown, Boveri & Co., who have also taken up the manufacture of Parsons steam turbines, have adhered to the original principle of blast governing; but oil under pressure is used instead of steam in the relay cylinder. The oil used is the same as that employed for lubricating purposes; thus, if the oil pressure fails, the relay piston cannot be lifted, and the turbine consequently stops. The use of oil instead of steam is claimed to have the further advantage that the working of the governor is practically independent of the steam pressure. The Westing-

house-Parsons turbine is also fitted with a blast governor, and the same applies to the Parsons type of turbine as manufactured by the Brush Electrical Engineering Company. Of course, in a turbine like the Parsons or Rateau, where there is a very large number of stages, the blast and throttle methods are the only two to choose from.

Those who are not in favor of the blast method argue that the times of admission of the steam into the turbine casing are far enough apart to allow the inside of the casing to be cooled by the drop in temperature between the puffs. Thus, as in the case of the steam engine, when there is a fresh admission the hot steam makes contact with the relatively cooler walls of the turbine casing and condensation takes place. But it is not easy to say to what extent this is true, and whether the efficiency of the turbine suffers to a great extent owing to this. With throttle governing there is naturally a loss owing to the velocities and pressure in the blades being different at light loads from the pressure and velocities for which the blades were designed, and further, the steam being throttled the pressure is reduced and the available energy derived from expansion is correspondingly less. The throttling process naturally dries the steam, or it may even superheat it; but this can contribute but little towards reducing the losses. It is, of course, a very simple matter to test a turbine at various loads with a throttle and blast governor, and so arrive at the relative efficiencies of each type.

As regards by-pass governing, it is obvious that economy in steam with such a method cannot be looked for, but the usefulness of the system is often great. Sometimes the by-pass valves are operated automatically and come into action as soon as the turbine is overloaded, whilst in other instances they are operated by hand. The practice of admitting steam at boiler pressure at the end of the first section—that is to say, considerably nearer the exhaust than usual—means poor economy, since the steam acts on many less blades; but by admitting steam nearer the exhaust the turbine can be made to develop considerably more power, and this is sometimes very important.

In the case of turbines driving dynamos in large power and lighting stations, for example, where it is important that the expenditure on plant should be kept down to a minimum, and where heavy "peak" loads have to be dealt with, the use of by-passes, although uneconomical, can have a favorable effect upon the amount of plant erected. When a set is under repair it is sometimes comforting to know that in the event of another unit coming to grief the remaining turbines are capable of 50 per cent., or even more, overload. But the trouble sometimes lies in the fact that, although the turbines are capable of these very big overloads, the generators are only rated for 25 per cent. overload for two hours and perhaps 50 per cent. for fifteen minutes, but speaking generally, the excess load is of short duration.

The Curtis system of governing by cutting-out nozzles is probably the most satisfactory method invented, although it is not suitable for turbines having a large number of stages, as in the Parsons and Rateau types. With this method the full initial pressure is maintained in all the nozzles that are opened, and there is consequently no throttling. Tests carried out on the Curtis turbines show that the steam consumption per horsepower or kilowatt hour does not vary greatly over a considerable range in load, and that this applies more particularly to the larger sizes. At the same time the Parsons, De Laval and other types have done very well in this respect, and, all things considered, it is somewhat difficult to judge which system of governing operates the best. The small-size De Laval turbines are fitted with a simple throttle governor, whilst the larger sizes, in the neighborhood of 200 kilowatts, governing is effected by cutting-out nozzles. Needless to say, the latter method gives much better results, but in small turbines where the total quantity of steam used is small, it is scarcely worth while complicating the machine with the necessary gear for varying the number of nozzles. In Germany, from which country most things intricate emanate, a system of governing has been devised in which adjustable slides are arranged in the various stages in front of the inlet openings of

the guide-vane wreaths, and judging from test figures, the results obtained with the system are highly satisfactory. Such a method of governing, however, is clearly not applicable to turbines having a large number of pressure stages, such as the Parsons and Rateau types.—"The Engineer."

CLEARANCE IN STEAM-ENGINE CYLINDERS.

It has long been known that maximum economy can only be secured in the commercial steam engine by complying with three conditions. First, the range of expansion must be large; secondly, the steam must be superheated; and, thirdly, the clearance must be as small as possible. Engines constructed to put these conditions in practice will give an indicated horsepower for between 9 pounds and 10 pounds of steam per hour. With a fairly good boiler this means 1 horsepower for 1 pound of good coal per hour. For the moment we shall not refer further to ratio of expansion or superheating; we shall deal only with clearance. It is certain that unless it is small economy cannot be had; but, curiously enough, it has never yet been definitely settled why. In other words, the relation between clearance and economy is apparently an unknown quantity.

It must be premised that we are dealing now with the steam engine as it is in practice, but we cannot avoid writing of steam as it is in theory. It will be found, then, that according to one theory there ought not to be any loss, while according to another theory a loss is certain. Rankine held that this loss was due to an alteration in the ratio of expansion. That is to say, if we have no clearance and expand a pound of steam through a given range, we shall have one result. If, now, there is a clearance of, say, 10 per cent., then we shall use 10 per cent. more steam, and we shall have a higher pressure at the point of release. To put it another way, instead of using a pound of steam to get a stroke of 1 foot we shall use 1.1 pounds. There is, however, a certain amount of compensation in the fact that

the average pressure throughout the stroke will be higher. But he says the trouble should be got over by taking care that the clearance space is always filled up to boiler pressure by compression. D. K. Clark reasons on somewhat different lines. He takes the weight of steam as a constant, and points out that if there is clearance, as it then occupies more space than it would if there was no clearance, the average pressure must be less with clearance than without it. But this condition can only exist on paper. The average pressure will in practice be rather greater with clearance than without it. The reasoning on either hypothesis is quite sound until we introduce another factor which complicates the whole question.

It is easily proved that no work can be done by an elastic fluid without change of volume, because, without change of volume, there can be no metamorphosis of energy. But the steam entering a cylinder during the admission period does not change in volume *qua* the engine. The work done during this period takes place in the boiler, and consists in raising the temperature of the feed water to the boiling point, and converting the water into steam. The steam contained in the pipe coupling the engine to the boiler may be regarded as a solid bar which is being continuously pushed out of the boiler. Lengths of this bar are severed stroke by stroke by the cut-off valve; and they are then supposed to increase in volume and to do work; but the whole process of the conversion of heat into work must be regarded as divided into two distinct operations, one of which takes place in the boiler and the other by expansion in cylinder.

If, now, the steam which fills a clearance space does not help to propel the piston, what becomes of the boiler-developed energy which it represents? To this question there are several answers, none of them conclusive. The general theory is that steam expanding as it does into the clearance space when the port is opened without doing work becomes superheated. It is by no means clear that this explanation will apply; the conditions are far too complex in practice. We have initial condensation to consider; the effects of the rush of steam; of its

loss of momentum, of the sudden rise of pressure from that of the exhaust in the cylinder to that of the boiler, are unknown quantities. In point of fact, although it is very easy to waste heat energy, it is very difficult sometimes to determine in what way the waste has occurred. If a gas under pressure is permitted to flow into a vessel in which a vacuum has been established, it is a fundamental thermo-dynamic proposition that no loss of energy can ensue provided the vessel is non-conducting. If it is not non-conducting, then the amount of loss will depend on the specific heat and weight of the vessel. Applying this law to clearance space, it is evident that no loss can take place, apart altogether from initial condensation, which is not special to clearance space, and may be disregarded for the moment. It follows, if this be true, that the comparisons between an engine with clearance and one without it is really a matter of the true and apparent ratio of expansion, and of nothing else. D. K. Clark says: "The clearance, or free space, between the piston when at the beginning of a stroke and the slide valve is filled with steam of the initial pressure at the commencement of each stroke; and this padding, as it may be called, does no work directly, and is entirely non-effective in non-expansive engines. But in expansion working cylinders the clearance steam does its proper quota of work in conjunction with the other steam during the period of expansion." In a subsequent table of the work which can be done under various conditions by a pound of steam, he gives in parallel columns the real and apparent ratios of expansion for cut-offs taking place at stated percentages of the stroke, on the assumption that the clearance is 7 per cent. of the cylinder capacity. Thus when the cut-off takes places at quarter stroke, instead of the expansion being fourfold it is 3.35.

In the passage just quoted we see that no attempt is made to explain what has become of the work which was undoubtedly in 7 per cent. of the steam when the engine is not expansive, yet must have been wasted on something. As the case is stated, however, it seems to be clear that, as we have said, clearance is only inimical to economy because it reduces the

amount of expansion. It ought to suffice, therefore, to cut off earlier and so get the required ratio of expansion, to cancel the loss due to clearance. No attempt seems to be made by any writer to dispute this view. It is, indeed, impossible to do it, and hold at the same time that the Joule double-bottle experiment is conclusive. This being so, the reason why in practice clearance is a source of great loss remains without any satisfactory explanation; and the difficulty in practice is enhanced by the fact that the evil effects of clearance do not appear to be diminished by compression. Let us, for the sake of argument, however, concede that the average pressure for a given weight of steam is diminished because more space has to be filled. Even then it will be found that any conceivable loss caused in this way will not be sufficient to explain the difference which can be brought about in the eonomic performance of an engine by apparently small differences in the percentage of clearance. We ourselves reject theoretical thermodynamic explanations as unsatisfactory. We hold that the value of small clearance must be sought in quite another direction. It is in the actual construction of the practical engine. Some of the most economical steam engines in the world have infinitesimal clearance spaces, because the distribution valves work directly on, or are seated directly in, the cylinder covers. There are no passages of any kind for the steam to run through, and so the metallic surface on which initial condensation can take place is minimized; and, further, the steam is presented in masses, so to speak, to the metal, instead of having to flow through narrow and tortuous passages, which seem specially contrived to bring every portion of the steam into contact with every portion of the metal, and in which passages steam can lie from the time it enters the engine until the exhaust port opens and it is pushed out, every facility being thus provided for its condensation. It is, of course, impossible to say whereabouts most steam is condensed in a cylinder. We have no doubt that it is a maximum in long passages and ports. The few experiments which have been made to test the effect of clearance have so far been inconclusive. There

can be no doubt, however, that in practice it is impossible to get high economy if the clearance is large. It is to be noticed, however, that all very economical engines are of the finest workmanship, and this circumstance should be kept in mind when we are drawing comparisons. It is quite possible that a result due to one cause may be attributed to another. No theory now extant will suffice to account for the results obtained in practice, and it is, therefore, legitimate to seek for further explanations. We believe that what we have suggested above will be found to have a good deal of weight; none the less, the whole question is worth more investigation than it has yet received.—"The Engineer."

AIR IN MARINE CONDENSERS.

From the days of James Watt onwards it has been known that air gets into condensers and vitiates the vacuum. Hence the name "air pump." But until Mr. Parsons discovered that an extremely high vacuum must be maintained in his condensers if the efficiency of his turbines was to be pronounced, no one attached very much importance to the part played by air. So long as the injection or jet condenser was used no steps could be taken to keep air out. It got in dissolved in the condensing water. Yet even in this case it was recognized that air in a cylinder did harm, and Cornish engines had their main stuffing boxes fitted with lantern brasses, supplied with water, to prevent the influx of air when the engine was "going out of doors." In 1862 the surface condenser began its life. We do not forget Hall or Craddock or, indeed, several other inventors; but the surface condenser as an adjunct to the marine engine dates no further back than 1862, and its invariable use goes no further back than the introduction of the compound marine engine. From that time onward—until, in fact, three or four years ago—interest centered in constructive details, the means of avoiding corrosion, preventing the slipping of tubes, the best method of packing tubes, and so on. If, for instance,

we turn to any standard book on the marine engine, we see that little account is taken of the influence of air in a condenser. Thus we find in Seaton's "Treatise on the Marine Engine," 1895 edition, a long chapter on air pumps, which begins: "Air pump.—The function of this pump in all condensers is to abstract the water condensed and the air which was originally contained in the water when it entered the boiler, and in jet condensers it pumps out in addition the water of condensation and the air which it contained." In dealing with the efficiency of the surface condenser neither Seaton nor any other writer of ten or a dozen years ago had a syllable to say about the part really played by air in a condenser. The idea was that it did mischief because it could not be condensed, and so kept up the pressure. It has remained for Mr. D. B. Morrison and a few other engineers to set forth the true function of air in the surface condenser. It does harm not so much because it cannot be condensed as because it impairs the efficiency of the tubes by preventing the transfer of heat from the steam to them.

The process of condensation, simple as it appears, is really very obscure, and is not fully understood. In a steam-engine cylinder one-fourth of all the steam entering will be initially condensed by a few square feet of fairly hot surface. To condense the remaining three-fourths in a surface condenser will require half as many square feet as there are of heating surface in the boiler, and a very large quantity of cold water. If it is argued that the steam is delivered intermittently to the cylinder, it may be answered that it is also delivered intermittently to the condenser. It has at all events been ascertained that if water is allowed to accumulate on the surface of the condenser tubes at the steam side much mischief is done and the efficiency of the condenser is reduced. But besides this, air plays its mischievous part, and precisely how is a matter still undergoing investigation. In the old anti-turbine days marine engineers were quite content with a 26-inch vacuum. Indeed, more was scarcely regarded as desirable. "It is the last straw that breaks the camel's back"—the last inch or so

of vacuum is terribly hard to get. It means more, much more, condensing water; an extra stress on the pumps, and cooler feed water. But the demands of the turbine are imperative; the last inch is as important as the first 27 inches, and it must be had.

On the 17th February Mr. Morrison read a paper before the Liverpool Engineering Society on "The Presence of Air in Condensers," which may be regarded as the latest word on the subject. He illustrated a very simple apparatus for ascertaining the amount of air in any sample of water; and he showed that air will not leave the water until it is close to the boiling point. In modern steamships there is usually a float-controlled feed pump, and with this the amount of air finding its way in is negligible. "Therefore practically all the air which passes through a condenser results from the leakage of joints and glands subjected to pressures below atmospheric. In marine practice, even when the condenser is connected to the main engines only, the amount of air passing the low-pressure piston-rod glands of a reciprocating engine may be very considerable, but when deck and other auxiliaries are discharged into the main condenser, the quantity is invariably so great that the condenser is always in a condition of having more or less of its condensing surface air-drowned and inefficient."

The words "air-drowned" require some explanation; it can scarcely be said that it is furnished by the following passage: "When exhaust steam enters a condenser the percentage of air to steam is so small that the temperature of the mixture corresponds practically to that of saturated steam; but in its downward flow steam is condensed and the air richness of the remainder increases until it reaches a maximum at the air-pump suction. But the greater the air richness the less the temperature, so unless the air pump is able to prevent an accumulation of highly aërated vapor in the lower part of the condenser the tubes in that part become air-drowned, and practically useless for condensation purposes, this action being intensified with increasing temperatures of circulating

water. It follows that the nearer the temperature of the vapor at the air-pump suction approaches that at the condenser top the more efficient will be the condensing surface, provided the design of condenser is such as will promote the gradual densification of the air by preventing short-circuiting of the steam."

No doubt Mr. Morrison had a very clear picture in his own mind of what takes place inside a condenser. We find it difficult to understand how stratification can occur. It is true that air is heavier than steam. This appears to be the only reason why highly aërated vapor should accumulate in the lower part of the condenser. So far as our own experience goes, there is never any difference between the pressure in one part of a condenser and another; indeed, so rapid is the destruction of the steam, that with a good condenser the smallest flickering of the hand of the vacuum gauge when the exhaust valve opens is the most that can be detected even with large paddle engines running at 25 to 30 revolutions per minute only. For the rest, the hotter the air is the larger will be the space it occupies. It should therefore be kept at as low a temperature as possible, if not in the condenser then in the air pump; and the latest refinement in condensation consists in cooling the air pump. A diaphragm or tray is fitted in the lower part of the condenser. This catches most of the condensed water, which is drawn off by a water pump. The air pump is placed in a tank of its own in which it is submerged. The water level is fixed by a float, and the arrangement is such that this water is circulated through the lower part of the condenser, and has a temperature not much above that of the condensing water. The air, of course, escapes from the surface of the water in the tank in the usual way. We quite agree with Mr. Morrison that the mere addition of tube surface will not improve a bad condenser, and that condenser is the worst in which most air can accumulate. In torpedo boats condensations at the rate of 30 pounds of steam per square foot per hour are reckoned upon, because care is taken that the air

pump shall, to use Mr. Morrison's words, "dominate the condenser."

It is not necessary here to say anything about Mr. Parsons' auxiliary air pump, or the various inventions now before the world for improving the efficiency of condensing surface. The phenomena of the transfer of heat from steam to water in a condenser deserves that thorough investigation which it has not yet received. Not only is the subject full of interest, but that interest is exceedingly practical. Results may be expected, indeed, of considerable commercial value.—"The Engineer."

THE LEBLANC EJECTOR CONDENSER.

The importance of obtaining high vacuum with turbine machinery has caused a great amount of attention to be paid to the subject of condensing plant during the last few years. Attempts at improvement have practically all been confined to modifications of the ordinary surface condenser, and probably there now remains little to be accomplished along these lines. Since the introduction of the Edwards pump, now almost universal, the question of the air pump has hardly been seriously attacked. At any rate, the type has held an unchallenged position for many years. It possesses, however, some defects inseparable from reciprocating machinery, which become the more evident the higher the degree of vacuum demanded. Hence M. Maurice Leblanc sought to devise a rotary pump which should fulfil the required functions, and we illustrate latest developments of the Leblanc condensing apparatus, as made by the British Westinghouse Company in this country.

Figs. 1 and 2 show the general arrangement of Leblanc air pumps as applied to an ordinary surface-condensing plant. A motor by the side of the condenser drives a centrifugal circulating pump coupled to one end of the armature shaft, and a rotary air pump at the other. Figs. 3 and 4 show sections through the air pump, and Fig. 5 the general appearance of a

Fig. 1.

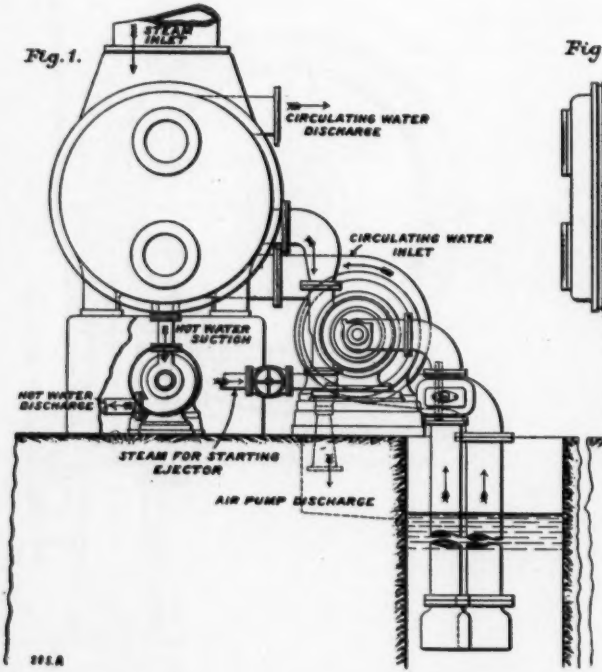
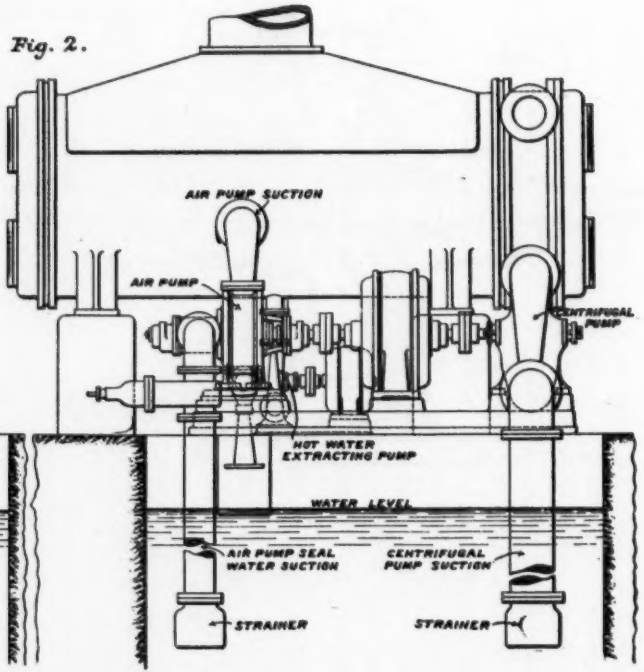

STEAM INLET
CIRCULATING WATER DISCHARGE
CIRCULATING WATER INLET
HOT WATER SUCTION
HOT WATER DISCHARGE
STEAM FOR STARTING EJECTOR
AIR PUMP DISCHARGE

Fig. 2.


AIR PUMP SUCTION
AIR PUMP
CENTRIFUGAL PUMP
HOT WATER EXTRACTING PUMP
WATER LEVEL
AIR PUMP SEAL WATER SUCTION
CENTRIFUGAL PUMP SUCTION
STRAINER
STRAINER

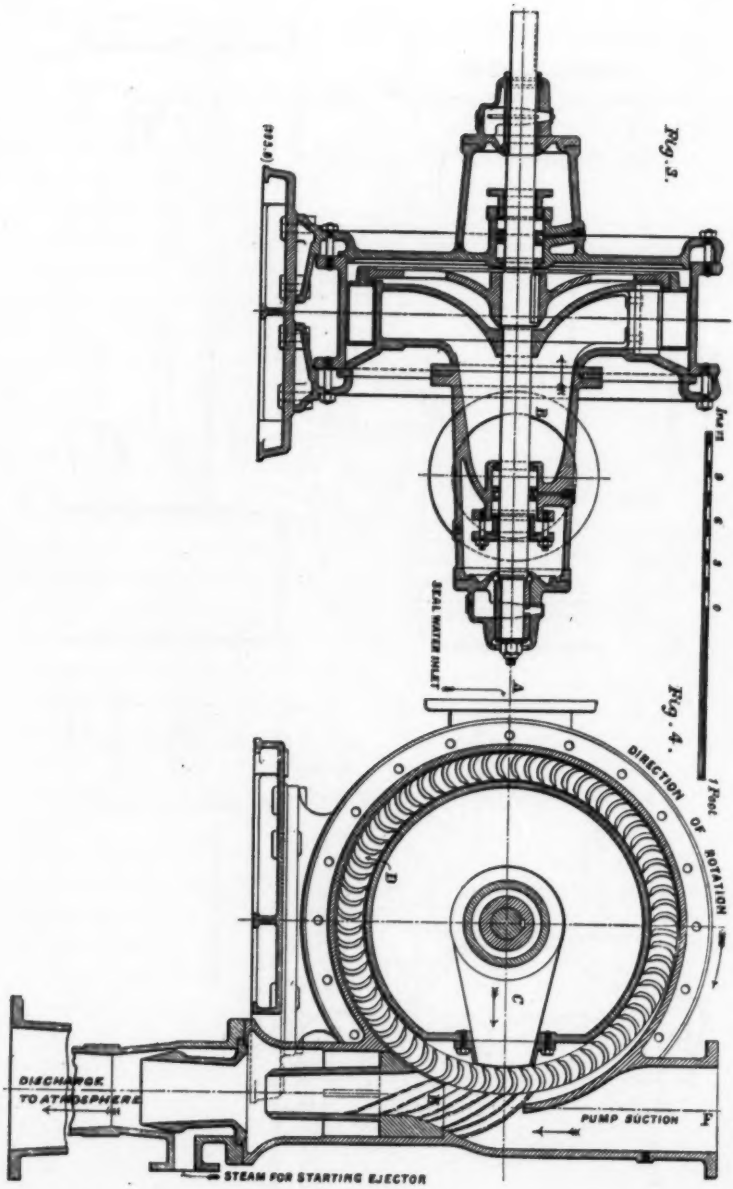


Fig. 3.
SEAL WATER INLET
Fig. 4.
DIRECTION OF ROTATION
DISCHARGE TO ATMOSPHERE
PUMP SUCTION
STEAM FOR STARTING EJECTOR

motor coupled to two similar pumps. Referring to Figs. 2, 3 and 4, water from a tank or reservoir is drawn up by the air pump and enters the latter by the branch A, whence it flows into the central chamber B. The only outlet from this chamber is through a horizontal nozzle by way of the port C. Fig. 3 is a bastard view, in that the outlet is made to appear at the top in order that it may be shown, whereas Fig. 4 shows it in its true position. The impeller of the pump is keyed to the shaft, and its rim is composed of a bronze ring furnished with blades. These blades are of cast bronze, and are cast into the

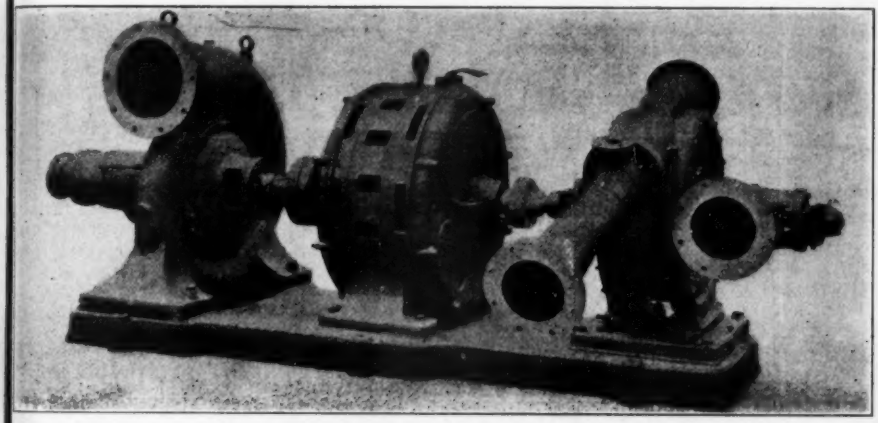
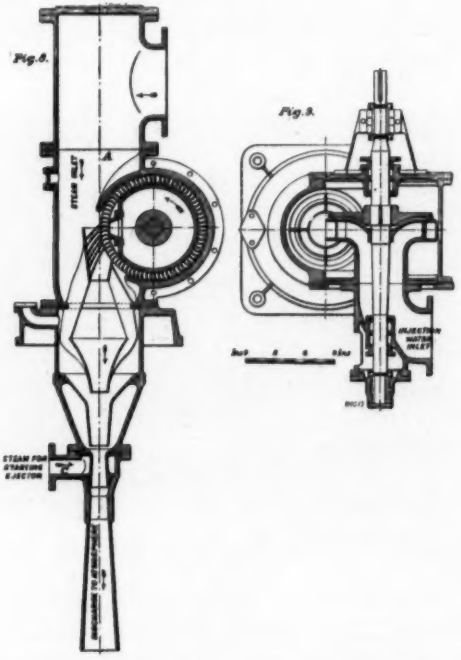


Fig. 5.—Motor-Driven Circulating Pump and Air Pump.

flanges which hold them, so that the completed ring is in one piece of bronze. The bladed ring D runs in an annulus between the walls of the central chamber and of the casing. As the blades pass the orifice of the central chamber they pick up the water and project it outwards in thin sheets, as indicated at E. Air from the condenser, entering by the branch F, is entrained by the sheets of water, and discharged with them against the pressure of the atmosphere. The water makes an effective seal, so that it is possible for the highest vacuum to be maintained behind it. An ejector nozzle will be seen in

the discharge pipe. This is for starting the pump. The motor is started, and live steam is turned on through the branch at the side of the discharge pipe. In a few moments a sufficient vacuum is created to enable the air pump to pick up its water and commence working. Steam is then turned off. The sealing water leaves the pump at practically the same temperature at which it entered, as it takes no part in the work



of condensation. Hence it may be used over and over again, or may be passed through the condenser subsequently.

Sometimes for small powers an ejector condenser is advisable on the grounds of cheapness and simplicity. The Leblanc air pump fulfils this function well, the arrangement being illustrated in Figs. 8 and 9. The Leblanc pump is built into the side of a vertical pipe, a little below the exhaust-pipe connec-

tion. It delivers water in the usual way, but in this case the water fulfils the double function of condensing and sweeping out the air. The sheets of water are thrown against the interior of a cone, and passing down through subsequent cones, and finally through the narrow throat of the discharge pipe, deliver the air against atmospheric pressure. As in the other cases, a steam ejector is fitted in the throat of the discharge pipe to start the apparatus.—"Engineering."

THE PRACTICAL EFFICIENCY OF HEAT-INSULATING MATERIALS.

By CHARLES R. DARLING, A. R. C. Sc. I., Wh. Ex., F. I. C., Etc.

In a previous number of "Engineering" (December 6, 1907) the author described a laboratory test for heat-insulating materials, designed to obtain results which would apply under actual working conditions. In this test a metal vessel, containing an electric heater, is lagged with the material, and the current adjusted until a steady internal temperature is attained; whereupon the heat escaping is measured by the watts consumed in the heater. The arrangement is such that any internal temperature between 500 degrees and 250 degrees Fahrenheit may be obtained, thus enabling a steam-pipe covering to be tested at its working temperature. It is proposed in the present article to give a brief summary of the results obtained during a long series of tests on commercial laggings, and to indicate some of the more important conclusions derived from the experiments.

The materials tested were in all cases obtained directly from the makers, each variety being procured in four thicknesses, viz: $\frac{3}{4}$ inch, 1 inch, $1\frac{1}{2}$ inches, and 2 inches, and, in some cases, six or eight thicknesses, ranging from $\frac{1}{2}$ inch to 2 inches, were tested, the loss of heat through each thickness being noted for a number of temperatures. As all the materials examined were intended as steam-pipe or boiler coverings, the outer surfaces were exposed to convection and radiation in the test, so as to imitate practical conditions as far as possible. (It

may be pointed out here that conductivity tests, in which the material is placed between two surfaces, give results which apply only to enclosed laggings—as used in cold stores, &c.—the loss of heat through a lagging with exposed surface being upwards of 10 per cent. greater.) All the samples were thoroughly dried before being fixed in position, and readings were not commenced until the specimen had been kept for two hours at the highest temperature embraced by the test. It was found, in some cases, that on prolonged heating at high temperatures the insulating power gradually improved, probably owing to the expulsion of normally-retained moisture; but in order to obtain a fair estimate of the insulating value under working conditions, in which a plant is in operation twelve hours out of twenty-four, a preliminary heating of two hours was decided on, as this gave the average figure. Difficulty was experienced in obtaining concordant results with some of the materials, owing to the separate thicknesses not being of the same density. In one instance the 1½-inch specimen proved superior to the 2-inch; whilst in another case, in which the samples were supplied in duplicate, equal thicknesses differed from each other to the extent of 10 per cent. of the total heat escaping. On the other hand, some of the laggings were quite satisfactory in this respect, and gave consistent results.

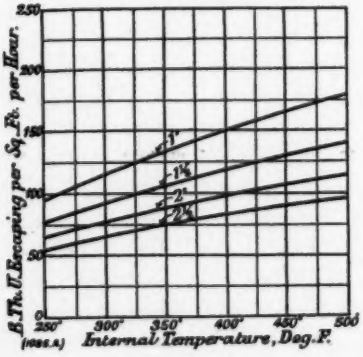
The figures obtained with two typical laggings are shown graphically in Figs. 1 and 2, the former of which refers to a sectional covering composed of 85 per cent. of light magnesium carbonate and 15 per cent. of white asbestos fiber; and the latter to a "plastic" composition, containing vegetable fiber and several other ingredients, and intended to be applied to the surface with a trowel. The heat escaping in each case is expressed as B.T.U. per square foot of lagged surface per hour, and the figures refer to a surrounding atmospheric temperature of 70 degrees Fahrenheit. It will be observed that in both diagrams the curves diverge from a common point (70 degrees Fahrenheit), and that at low temperatures the saving occasioned by the use of a thicker lagging is much less marked than at high temperatures. In Fig. 1, for example, the su-

periority of 1½ inches thickness over 1 inch is represented by 17 B.T.U. per square foot per hour at 250 degrees Fahrenheit, whilst at 450 degrees Fahrenheit the extra saving is 36 B.T.U. It will also be noticed that the additional saving effected by successive half-inches in the thickness is a diminishing quantity; thus, in Fig. 1, at 400 degrees Fahrenheit, the 1½-inch is 30 B.T.U. superior to the 1-inch; the 2-inch better by 20 B.T.U. than the 1½-inch; whilst the 2½-inch is only 14 B.T.U. superior to the 2-inch. Each successive layer beyond the 2½-inch would show a decreased effect, and evidently a stage must be reached at which the cost of an extra layer will exceed its additional saving power. The economic limit for a given lagging can readily be obtained from the curves, when the working temperature and probable duration of the lagging are known. As an example, suppose it were desired to know whether 2½ inches magnesia would be more economical than 2 inches, when used to lag pipes conveying steam at 400 degrees Fahrenheit, for an estimated period of 30,000 working hours. Reference to Fig. 1 shows that at 400 degrees Fahrenheit the 2½-inch thickness saves 14 B.T.U. per square foot per hour more than the 2-inch; hence the extra saving in 30,000 hours would be 420,000 B.T.U. Taking 1d. as the equivalent of 100,000 B.T.U., the superiority of the 2½-inch lagging is represented by 4.2d. in the given period. If the difference in cost of the two thicknesses be less than 4.2d. per square foot, it will be cheaper to use the thicker material; if greater than 4.2d., the thinner lagging will be more economical. It is evident that the correct thickness to use, under any given circumstances, can be derived in a similar manner. At present engineers are largely guided by custom with regard to the thickness of lagging; and whilst, in the main, the thicknesses adopted are fairly suitable, there are many instances in which hot surfaces are absurdly underlagged or overlagged, owing to the absence of data from which the correct thickness could be determined. By comparing the curves for a number of materials, the most economic to use for a given purpose can be deduced; and it will generally be found that for high-tem-

perature work the superior insulators, in spite of the greater prime cost, are cheaper in the end than inferior types; whilst at low temperatures the contrary may prove true. Each case, evidently, must be decided on its own merits, the factors being prime cost, working temperature and estimated duration.

A comparison of the two sets of curves (Figs. 1 and 2) shows a noteworthy distinction In Fig. 1 the curvatures become less as the temperature increases, indicating that the insulating power improves relatively to the temperature, whereas in Fig. 2 the curvatures increase, showing a falling-off in insulating value as the temperature rises. The improve-

Fig.1. EFFICIENCY CURVES FOR MAGNESIA LAGGING.

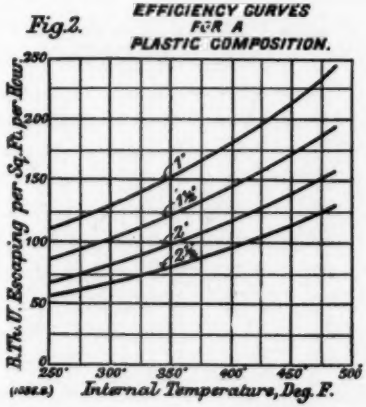


ment observed with magnesia is probably due to the expulsion of moisture—and possibly excess of air also—as the temperature rises, and might not be observed in a test in which the material was enclosed between two surfaces, as the escape of moisture and air would then largely be prevented. With an exposed surface, however, the improvement mentioned is noticed; tests on this material made at the National Physical Laboratory and elsewhere having shown a similar result. This point is of great importance in selecting a lagging for pipes conveying highly superheated steam, for which purpose a material giving curves such as those in Fig. 1 is to be preferred—

other things being equal—to one which behaves in the opposite sense. Blue asbestos, when used as fiber, resembles magnesia in this respect, and also, according to some observers, slag-wool. Practical experience of the use of these materials for high-temperature work confirms these observations, which explain why the superiority of some laggings over others is more manifest at higher temperatures.

The curves obtained with other materials resemble either Fig. 1 or Fig. 2 in general character, blue asbestos fiber giving results similar to Fig. 1, but with slightly higher values for the escaping heat; whilst most other laggings afford curves of the

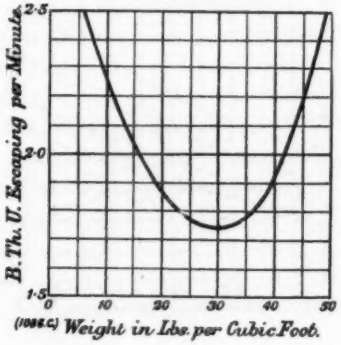
Fig. 2.



type shown in Fig. 2, although varying considerably with respect to the quantity of heat escaping. Mica and white asbestos sectional coverings give figures indicating a less escape of heat than is shown in Fig. 2, which may be taken as giving the average result for a good plastic composition. Inferior plastic materials show a much greater heat loss than Fig. 2, but, owing to the great differences in the densities of what purported to be the same laggings, the author is unable at present to give comparative figures for a number of specific compositions, the work on which is still in hand. It is evident that a comprehensive series of tests on the lines described

is necessary in order to form a correct judgment of the thermal properties of a lagging, and if a comparison be made at one temperature only, a correct inference can only be drawn by conducting the test at the actual working temperature. It is quite possible for a material at low temperatures to give results inferior to those obtained with another material, and yet to be superior to the latter at high temperatures. The lack of agreement observed in the results of tests on the same materials by different workers is partly to be ascribed to the use of different working temperatures, and partly to variations in the density of the materials tested.

Fig. 3. ESCAPE OF HEAT THROUGH SLAG-WOOL OF VARYING DENSITY.



The effect of density on insulating value has not hitherto received the attention it deserves. Inasmuch as a porous solid is a better insulator than an air space, or the same solid in non-porous form, it follows that in all cases a definite ratio of air space to solid must exist, at which the insulating power is a maximum. An illustration of this is seen in Fig. 3, which represents the heat passing through layers of slag-wool of equal thickness, and at the same internal temperature, but of different density, the escaping heat being that actually observed in the experiments, and not referred to any given area of

surface. It will be seen that the maximum efficiency is possessed by a layer of density nearly 0.5, or 30 pounds per cubic foot, and the best results would be secured by using slag-wool of this density. Many laggings do not permit of wide variations being made in the density; but wherever possible it would be a distinct advantage to determine the best density, and to make up the lagging accordingly. A striking practical example of the effect of density is to be found in the case of kieselguhr, which, in the native state, is one of the best mineral insulators. When injudiciously blended with binding material, however, the porous character of the kieselguhr is largely destroyed, and the resulting lagging is relatively an inferior insulator. It is no exaggeration to state that the correct manufacture of a lagging is of equal importance to its composition.

In conclusion, it may be stated that the results obtained by the author, although in many respects incomplete, show the necessity of conducting a complete set of tests on a material in order to decide the correct conditions of its use, and that these results may be arrived at when material of uniform density is obtainable. In this respect the method adopted marks an advance on those previously in use for testing laggings under working conditions, and has already been found of practical service. It must always be borne in mind, however, that thermal tests alone are insufficient to decide the choice of a lagging, and that considerations of mechanical strength, surface temperature, resistance to moisture, &c., are often of equal importance.—"Engineering."

SOMETHING ABOUT THE CORROSION OF IRON.

By Dr. Wm. H. Walker.

The material which we usually term iron is unique in that it is never in reality the pure metal as is, for example, the case with copper, but is a mixture of iron and certain alloys of iron with other elements with which it unites during the

process of its manufacture. These alloys or compounds, moreover, when present even to a very minute extent, have a controlling effect upon the physical properties of the resultant product. When the iron is first reduced from its ore these impurities are present in relatively large quantities, and are removed by refining processes which, if carried on with the iron in a plastic condition, results in what we term wrought iron, while if during the process of refining the mass is held in a molten condition, the material known as steel is produced. The iron of our forefathers was all made by the first method, and it cannot be denied that such material resisted corrosion, or rusting, as this phenomenon is usually termed, better than much of our modern steel; but inasmuch as we have now offered on the market both good and bad iron and good and bad steel, the long discussion which has recently been carried on as to the kind of metal which resists corrosion better is without any value whatever. It may be said, that if all iron and steel withstood corrosion as does *the best* iron and steel there would be little interest in a further discussion of the subject.

The factors which determine whether a given sample of iron or steel will corrode rapidly or slowly, and the conditions upon which corrosion in general depend, may best be understood by first considering this phenomenon from a theoretical point of view. Every metal when placed in contact with water has a certain tendency to pass into solution. This tendency to escape is known as solution pressure, and varies with the material. In a metal like sodium this pressure is so great that instant solution occurs when the metal comes in contact with water, while in the metal platinum the solution pressure is negative in value. In passing into solution the metallic particles change from an atomic condition to the ionic condition, and in so doing assume a positive charge of electricity, leaving the metal itself negatively charged. There will thus be formed an electrolytic double layer and the action will cease. If, however, there be present in the water another ion which has a less solution pressure than the metal under consideration

such ion will pass from its ionic condition to its atomic condition and separate out upon the first metal, giving up its positive charge of electricity thereto. An electric current will therefore pass through the solution from one point on the metal to the other, destroying the electrolytic double layer and causing the solvent action of the metal to be continued.

When a piece of iron is immersed in a solution of copper sulphate, for example, an analogous reaction takes place in which the iron passes into solution, metallic copper plates out upon the metallic iron, and an electric current flows from those places where the iron is passing into solution to those portions where the copper is plating out and through this copper to the iron again, although these points may be infinitely near together. If, now, the bar of iron be placed in water instead of the copper sulphate solution a perfectly analogous reaction takes place, except that instead of copper separating out on the iron we now have one of the ions of which water itself is composed, viz: *hydrogen* separating out on the iron and an electric current flowing as before. Water is composed, or breaks up into, the two ions, hydrogen, which carries the positive charge, and what is termed the hydroxyl ion, which carries an equivalent negative charge. It happens that there exists a very delicate test for the presence of the iron ion from the fact that potassium ferri-cyanide solution immediately forms therewith a bright blue color; while, on the other hand, there is an organic substance known as phenol phthalien, which becomes intensely red in the presence of hydroxyl ions. If, therefore, a piece of iron be placed in a water solution containing potassium ferri-cyanide and phenol phthalien, and which, to prevent convection currents, be thickened with a little ordinary gelatine, there should be formed a bright blue color at those points on the iron bar at which the iron is passing into solution, and a bright red color on those portions on which the hydrogen is plating out, leaving in the solution at that place an equivalent number of hydroxyl ions. These conditions are easily realized, and a number of such specimens were shown. The fact that iron in the

presence of a phenol phthalein solution colored the same red was first noted by Dr. A. S. Cushman, of Washington, D. C., and used by him to prove the presence of ferrous hydroxide in solution. This observation in the hands of Dr. Walker has resulted in this splendid and altogether convincing proof of this first and important step in the corrosion of iron.

According to this conception, therefore, if a piece of iron be placed in water in the absence of air or other foreign material, iron should continue to pass into solution and hydrogen should continue to plate out in the form of gas, but this is not in fact the case. This solvent action in water proceeds but to a very slight extent and then stops, while when iron is placed in a copper sulphate solution under similar conditions the action is continuous. In the case of the copper, which plates out on the iron, we have a good conductor for the electric current which must pass from one point of the iron through the solution and the copper back to the iron again. The film of hydrogen which plates out on the iron in the second case is, however, not a conductor of electricity, but an insulator, and until such film of hydrogen be removed the current cannot pass and the action cannot continue. This hydrogen film may be removed by any reagent which will unite with hydrogen, such, for example, as the oxygen of the air. If, therefore, oxygen be introduced into the water in which the iron has started to dissolve, this hydrogen film will be rapidly destroyed, the electric current will pass and the iron will continue to dissolve or corrode.

A second function of the oxygen, and although only incidental to corrosion, is in reality the more striking, is the conversion of the iron thus thrown into solution in the water into the red hydrous oxide of iron with which we are familiar as rust.

Having now a conception of the mechanism by which iron corrodes, some important conclusions may be drawn from a consideration of the factors involved. Since before the iron can form rust it must first pass into solution, and in so doing cause an electric current to flow from the iron at that point to

the iron at some other point, any circumstance which will aid the flow of this current will accelerate the solution of the iron. In other words, any differences in potential which may exist upon the surface of the iron will in itself cause a flow of electricity which will result in a solvent action on the iron. Such differences of potential inevitably result from a segregation or uneven distribution of any impurities which the iron or steel may contain. Hence we would expect that the speed of corrosion would increase, first, in accordance with the percentage of impurities present, and, second, in accordance with the care bestowed upon the iron or steel during its manufacture to prevent a segregation of these impurities. The simplest way, of course, to insure an absence of segregation is to eliminate altogether those materials not needed in the iron. It has been found that a steel made under such conditions that the total impurities are reduced to not over five one-thousandths of one per cent. resists corrosion to an extent equal to the iron of our forefathers. The steel companies have been slow to accept this general proposition, but it is gratifying to know that material of this purity may now be obtained on the open market.

Since iron can pass into solution only by assuming the ionic form, and having at the same time an equivalent amount of hydrogen pass from the ionic condition to the gaseous state, the speed of the reaction will depend upon the number of hydrogen ions present in the solution. If this number be enormously increased, as can be done by adding some strong acid, the reaction will be accelerated to such an extent that the hydrogen may be seen passing off the iron as a stream of bubbles. If this concentration of hydrogen ions be but slightly increased by the addition of such weak acids as the carbonic acid of the atmosphere, or the carbonic and sulphurous acids resulting from the combustion of most fuels, corrosion will again be accelerated, though to a less noticeable extent. The presence of any material, such as alum or magnesium chloride, when subjected to pressure, or other materials which by hydrolysis will form an acid, will also accelerate corrosion.

If, on the other hand, the hydrogen concentration be reduced the corrosion of the iron will also be decreased. Such a condition may be most easily realized by adding to the water a strong alkali, which, owing to the presence of a large number of hydroxyl ions, decreases thereby the number of the hydrogen ions. Iron will not rust, therefore, in a solution of soda ash, limewater, or any materials such as sodium phosphate, which when present in water will by hydrolysis decrease the number of hydrogen ions.

This fact has an important bearing upon the question which has been lately much discussed, as to whether concrete will protect iron from corrosion. Inasmuch as Portland cement when it "sets" or hardens liberates a large quantity of caustic lime, which is a strong alkali, and since good concrete is therefore saturated with this strong alkali, the answer to the question must be in the affirmative. Iron or steel will not corrode when imbedded in concrete. But caustic lime is soluble in water, and poorly made concrete is not impervious to water. Therefore if iron be imbedded in concrete through which water at any time be allowed to percolate, this calcium hydrate will be slowly but surely dissolved and washed away, and with it will disappear the inhibiting action of the concrete, and iron imbedded therein will surely rust. To insure absolute protection of the reinforcing members of concrete construction, therefore, such concrete must be sufficiently dense and carefully made to render it practically waterproof. This is not an impossible condition.

The solubility of calcium hydrate present in concrete has an important bearing upon the corrosion of certain of the steel members of the new Cambridge bridge, which has excited comment in some of the Boston papers within the last few weeks. The central portion of the structure, designed to accommodate the tracks of the Boston Elevated Railroad, may be still easily inspected. Between the ends of the I-beams, on which rest the heavy granite curb of the highway on either side, corrosion of the iron has already commenced, although the bridge has been in use but a few years, and, according to the newspapers,

has been twice painted. The cause of the trouble is found in the fact that water from the highway portion of the bridge has percolated down to the concrete and has leaked out between the joints in the supporting I-beams, carrying with it the soluble calcium hydrate. This strongly alkaline solution has saponified the linseed-oil paint used on the bridge, and when subsequently removed by rain corrosion has, as it must inevitably do, ensued. If there be steel members in the cement of the highway proper they will also corrode so soon as the lime of the concrete has been removed by the water.

Since corrosion can take place only when the hydrogen film is continually destroyed, it follows that any agency which will increase the speed of the reaction by which the hydrogen which plates out on the iron and the oxygen which dissolves in the water will unite, will in turn accelerate the corrosion. This union of hydrogen and oxygen takes place more rapidly on the surface of copper or lead or bronze than it does on iron. Hence if there be connected with the iron one of these metals, the hydrogen of the reaction will be separated upon it, and the iron will dissolve more rapidly on this account. Ordinary mill scale, which is magnetic oxide of iron, is also a substance on which hydrogen can be liberated more easily than on iron. Hence if there be a piece of iron, for example a boiler tube or an I-beam, covered with mill scale, and there are breaks in this scale, the iron at these breaks will corrode more rapidly because of the scale with which the breaks are surrounded. This is one cause for the pitting of boiler tubes, and has led the navy in many instances to remove all of this mill scale from their boiler tubes before being placed in position. The above phenomenon emphasizes the care which must be employed in structural work to prevent the contact of iron with such materials as bronze, lead or copper, unless they be electrically insulated therefrom, and the desirability of removing the mill scale before painting.

On the other hand, electrically connecting a piece of iron with a bronze or copper structure will protect the latter from corrosion, because, owing to the fact that the iron passes into

solution, hydrogen will be liberated upon the bronze or copper and its corrosion will thus be prevented. Employing the same reasoning, we find that any metal with a greater tendency to dissolve than iron, for example zinc, on which the hydrogen is liberated with more difficulty than on iron, should protect the iron at the expense of this other more positive metal. This is the explanation of the well-known use of zinc in boilers for the prevention of pitting, and it is but necessary to maintain an electrical connection between the zinc and the iron to insure the complete protection of the latter for a distance which depends upon the electrical conductivity, or purity, of the water in which the two are immersed. In perfectly pure water zinc will protect iron but for a distance of one-half inch or so; if salt, or any other electrolyte, be added thereto the area of protection is increased to twenty inches or more.

Since oxygen of the air is necessary to insure the continuous removal of the hydrogen film, it is obvious that if no oxygen be allowed to reach the iron through the water no corrosion can take place. This fact teaches us much regarding the corrosion of boiler shells and tubes. Pitting may be entirely avoided if the air be removed from the feed water before its introduction into the boiler. This may best be done by the employment of an open feed-water heater or, what is better still, a feed-water heater connected to the dry-vacuum pumps of the condenser. If such treatment is not possible the air may be removed from the feed water by drawing the water through a closed box containing scrap iron; the oxygen in the water is used up in corroding the scrap iron instead of the boiler tubes. Or, the oxygen in the water may be absorbed by feeding into the boiler with the water a very small quantity of an alkaline solution of a tannin material. Such a solution of alkaline tannate will break up under the pressure and temperature of the boiler, with a formation of a pyrogallate of the alkali, and this rapidly absorbs the oxygen. Soda ash, or other alkali, is, of course, useful, but not because of its effect upon the oxygen content, but because, as has already been explained, corrosion is inhibited by thus decreasing the hydrogen concentration of the water.

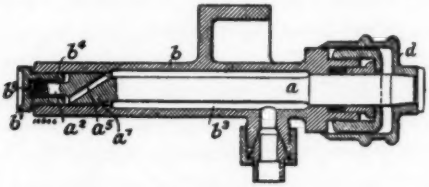
Tin is a metal which, like copper, accelerates the corrosion of iron by aiding in the oxidation of the hydrogen set free by the reaction. If, therefore, in the manufacture of the so-called tin plate, which is sheet iron or steel covered with a layer of metallic tin, there be imperfections or pinholes in the tin coating, these channels through the tin to the iron will become centers of corrosion which rapidly destroys the plate. It is impossible to detect these imperfections by a simple inspection, and hence users of tin plate have been unable to test the quality of their raw material from this point of view. Since the iron must pass into solution at these exposed points in the tin coating, their presence may be easily located by flowing upon the tin surface a solution of gelatine or glue in which is dissolved some potassium ferri-cyanide. When the jelly has stiffened the iron will pass into solution through the holes in the tin and, reacting with the ferri-cyanide, will leave a bright blue spot. Variations in the quality of tin plate may thus be easily detected.

The above are a few of the applications which have been made of the electrolytic theory of the corrosion of iron and steel. It is believed that our natural store of iron ore may be conserved and the durability of our iron and steel structures greatly increased by a further application of the knowledge thus made available.—"Bulletin of the Society of Arts, Massachusetts Institute of Technology."

LIQUID-FUEL BURNERS.

This invention of J. Dewrance and J. H. Rosenthal, London (July 14, 1908), relates to apparatus for spraying liquid fuel for burning in steam-generator and other furnaces. The apparatus comprises a spindle a formed with a screwed portion a^1 and with a cylindrical extension a^2 of smaller diameter than the spindle. The screwed portion a^1 works in a threaded portion of the casing b, while the extension a^2 is housed within a cap b^1 formed with a central orifice. The extremity of the

extension a^2 is formed with spiral passages, preferably after the fashion of a square-threaded screw, and closely fits that portion of the cap b^2 within which it is housed. The spiral passages are intersected by a number of longitudinal channels. The liquid fuel arriving, under considerable pressure, by way of the inlet *c*, passes by way of the annular passage b^3 and a diag-



onal passage or passages a^5 to the annular passage b^4, and thus reaches the spiral passages and longitudinal channels. A portion of the liquid is thus caused to flow spirally, whilst the remainder flows in a longitudinal direction. As these streams intersect one another at numerous points, the liquid becomes broken up to such an extent as to preserve its disintegrated or atomized condition, although subsequently caused to converge towards the central orifice, whence it issues in the form of a finely-divided spray, the angular dispersion of the jet being comparatively small. For the purpose of regulating the discharge of the sprayed liquid from the orifice the rear end of the spindle *a* is provided with a head *d*, whereby the former may be rotated and the space between the front extremity of the spindle and the inner face of the cap b^2 increased or diminished as desired, or the outlet closed.—"Engineering."

OIL FUEL RESULTS.

Except in the case of steamers built for carrying oil in bulk, or those trading to or from, or in the region of, the oil fields, it can scarcely be said that liquid fuel has made much headway in the merchant service. It is true that the evaporative efficiency, weight for weight, is from 33 to 70 per cent. better than coal, according to the calorific value of the latter, and

that the reduction in bunker capacity and in weight of fuel carried enables more cargo to be taken, or a smaller ship to be built for a given duty, or a higher speed to be realized for the load carried. In the case of the high-speed Cunarders, for instance, there might have been a possible reduction of between 2,000 and 3,000 tons in displacement, and this would have increased greatly the speed for the same power, or reduced the power for the same speed. But with oil in this country selling at 60s., as compared with anything from 10s. to 20s. per ton for coal, there is not much inducement to adopt the liquid-fuel system. Indeed, the great cost is carrying weight with economists, even at the admiralty, notwithstanding the great strategical and tactical advantages of oil fuel already well known.

There are other gains, however, in the reduction of the cost of burning the fuel—in the loading, trimming and stoking of fuel, in the smaller staff required, and in the repairs, etc., bill. It has been difficult to get exact comparative data of the financial results of burning oil and coal, apart from the calorific results. In the accompanying table there are given such comparative figures from three merchant ships in service

COMPARISON OF THE LABOR COST PER DAY OF BURNING COAL AND OIL IN THREE STEAMERS.

Example.	Fuel.	Boiler draft.	Tons of fuel used.	Number of firemen.	Wages of stokehold complement.			Victual-firemen, etc.			Repairs and maintenance.			Total cost.			Total cost per ton of fuel.	
1	2	3	4	5	6			7			8			9			10	
					£	s.	d.	£	s.	d.	£	s.	d.	£	s.	d.	s.	d.
1	Coal	Nat.	22½	11	1	9	4	0	11	0	0	5	5	2	5	9	2	0
	Oil	Nat.	15	3	0	8	0	0	3	0	0	0	9	0	11	9	0	9½
2	Coal	Fcd.	29½	14	1	17	4	0	14	0	0	10	1	3	1	5	2	1
	Oil	Fcd.	19½	3	0	8	0	0	3	0	0	1	0	0	12	0	0	7½
3	Coal	Fcd.	300	...	19	12	0	12	13	0	5	5	0	37	10	0	2	6
	Oil	Fcd.	200	12	2	0	0	0	18	0	0	7	6	5	5	6	0	4

There is given a comparison of the expenses in three different ships, the third a moderate-sized cargo carrier. Column 4 gives the consumption respectively in each ship for the same service, and it will be seen that the oil is about 33 per cent. less than the coal; in other words, two tons of oil give the same energy at the propeller as three tons of coal. The number of firemen (column 5) and the wages paid (column 6) in the smaller steamers when using oil fuel is less than one-fourth what they are when using coal; but with increased size of ship and greater power of machinery the gain is augmented. Thus the owner of the cargo steamer using oil pays in wages only one-fifth of the sum disbursed when burning coal. Oil does not need to be trimmed nor conveyed by men from distant bunkers. For very small cost in steam-pumping power it is automatically fed into the boiler, which is not the case with coal. The relative repair and maintenance costs given in column 8 are most significant: liquid fuel is a preservative of steel; as the boiler doors are never opened, as with coal burning, there are no such sudden variations in temperature as those which affect junctions of tubes with tube plates, and involve other troubles with coal-fired boilers. The net result is that oil in a steamer using 200 tons per day costs 4d. per ton to burn, against 2s. 6d. for coal. This does not include the first cost of the fuel or plant for separating water from the oil, heating, compressing and air-spraying the oil.—"Engineering," London.

THE NEW NAVAL ESTABLISHMENT AT ROSYTH.

The Admiralty have let the contract for the extensive dock works in connection with the new naval establishment at Rosyth, and Messrs. Easton Gibb & Son, Limited, Great George street, Westminster, will at once proceed with the undertaking, for the completion of which the Admiralty have specified seven years for the main works, and four and a half years for those in connection with torpedo and submarine craft. A bonus is to be paid for anticipation of this date.

The firm originated in Scotland about eighty years ago, and carried out a considerable part of the Great North of Scotland, Callander and Oban, and other Railways; and in the thirties they constructed the Glasgow Bridge over the Clyde. Some years ago the firm removed their headquarters to London, and have done some important work in England, the present head of the firm, Mr. Easton Gibb, being the son of the late Mr. Alexander Gibb, who was consulting engineer to the Great Northern of Scotland Railway Company. Amongst their later work is the Kew Bridge, docks at Cardiff, and railway works at Newcastle, while they have at present in hand extensions to Newport Docks.

The works at Rosyth, so far as authorized, will cost in all about 3½ millions sterling, and will include a large basin, entrance lock which may be used as a graving dock, an emergency entrance and exit in addition to the main channel, a drydock, boat slip, basin for submarine boats, and workshops for repairs to the Fleet. The total area covered is 1,184 acres of land with 285 acres of foreshore, and the site has a frontage to the Firth of over two miles. There is a naturally-protected anchorage, known as St. Margaret's Hope, and the dock works are situated to the east of this anchorage. The main channel is to be dredged to a depth of 36 feet at low water of spring tides, and at the bottom will be 1,050 feet in width. It will be cut at an angle to suit vessels steaming to and from the North Sea. In order to facilitate the guidance of ships into the entrance lock, and thence to the basin, there will be a pier 800 feet long in line with the channel. The emergency channel is a feature which corresponds with the arrangement at the new basin at Keyham; it is 110 feet wide, and will be used by vessels entering and leaving when the entrance lock is occupied as a graving dock.

This entrance lock is an important part of the works. It will have a length of 850 feet, a width at the entrance of 110 feet, and the depth of water over the sill at low water of ordinary spring tides will be 36 feet.

In addition to this lock there will be a graving dock project-

ing from the north wall of the basin. The length will be 750 feet, but provision will be made so that it may be easily and cheaply lengthened to 1,000 feet; the width will be 100 feet at the entrance, and the depth of water on the sill 36 feet. An intermediate caisson will be arranged for, so that the inner part of the dock may be used for a vessel 280 feet long, and the outer part for a ship 450 feet long. The main basin is approximately square, the width from north to south being 1,620 feet, and from east to west 1,500 feet, giving a total area of 52½ acres. From the western wall there will project a pier 600 feet long, which, while adding to the berthing accommodation, is primarily intended to assist in the canting of ships. The depth of water arranged for within the basin is 38 feet 8 inches, and the total length of the berthage is 7,100 feet. The south wall is to be extended towards the main channel by a river wall over 2,700 feet long, with 36 feet of water alongside. This is to be utilized for coaling purposes, and accommodation will be provided for storage of fuel.

To the east of the main basin there is the torpedo and submarine depot, the main feature here being the camber, 500 feet by 470 feet, with a depth at low water of ordinary spring tides of 15 feet, for the accommodation of twelve submarines and six destroyers. Alongside there will be a boat slip 260 feet long and 130 feet wide, about one-half of which is to be covered with light roofing. Further facilities for repairing these small craft will be afforded by a floating dock 250 feet long. The submarine depot, boat slip, pumping station and electric-power house, are to be completed in 4½ years.

The monolith system is to be adopted in constructing the dock walls, borings having shown that this method would be preferable. The plans arranged provide for concrete monoliths, about 43 feet square, which will be built up and sunk to the boulder clay by the usual procedure of first fitting a steel shoe and building over it concrete walls within shuttering. There will be four square compartments on the plan to facilitate the sinking.

On the landward side of the basin there are to be extensive

repairing shops, including a large boiler shop and smithy, with electric-light and power station, and the usual accessories—extensive stores, surgery, cranes, latrines and the other necessary equipments of a naval base will be added. Already there is railway communication with the Forth Bridge line, and much of the preliminary work has already been constructed, so that it is probable that active work will be in progress at an early date.

In addition to the usual equipment, such as bollards, &c., the contract includes a considerable amount of dredging.

Although the scheme as now approved is complete in itself, care has been taken so to arrange it that it can at any future time be extended without interfering with anything which is done now. Besides an additional basin, a number of docks of the largest size can be readily constructed, in addition to all the shore requirements for a first-class naval port.—"Engineering."

USE AND MISUSE OF GRAPHITE.

Graphite properly used is the engineer's friend, and there are numerous uses to which it can be put with very satisfactory results. It is good to mix with oil and use on the bolts and nuts of cylinder heads, steam-chest covers, pipe flanges, etc., because it makes the nuts work easy when one wishes to break a joint. If the nut and bolt of the rear handhole plate of a horizontal tubular boiler are well coated with graphite, and the bolt, nut and crab covered with a few handfuls of asbestos mortar, they will stand a fierce heat for months, and a wrench is all that will be needed to remove the plate.

Never plaster graphite and oil all over a gasket and then cuss because the packing slides out of the joint when the bolts are set up. Graphite one side of the gasket only. Put the gasket on the plate and graphite the exposed side. The gasket will then come off with the plate and leave the other surface clean and smooth. For this purpose the graphite should be mixed to a thick paste with cylinder oil.

Rod and valve packing for steam should be given a liberal coat of graphite when putting in place. The packing commonly used in the water end of a pump is made of several layers of heavy cotton cloth cemented together with a rubber compound, and it is nothing uncommon for the rings of packing to stick together so firmly, especially in hot water, that the expanding device is unable to set them out as they wear. A little graphite between the rings will prevent their sticking and the packing will run much longer without attention.

As a lubricant for steam cylinders and internal-combustion engine cylinders graphite is undoubtedly valuable, but the difficulties of feeding it discourage many from trying it. To get graphite into a cylinder all that is needed is a small cup with a straight, free outlet.

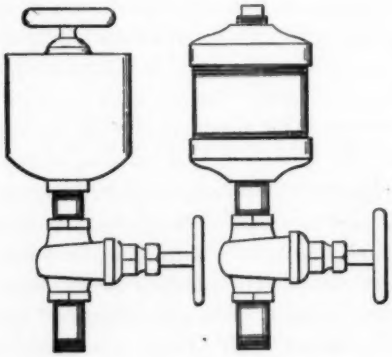


FIG. 1

FIG. 2

Fig. 1 shows how a cylinder-oil cup may be made over to feed graphite. Use a gate valve or plug cock, and attach the cup to the steam chest, or as near to it as possible. Fig. 2 shows how a cup may be made of pipe fittings, and, if carefully made of brass, it does not look bad.

Having attached the cup, put in about a teaspoonful of oil and graphite, mixed to the consistency of paint, close the cup and open the valve wide. In my plant I have a $7\frac{1}{2} \times 6$-inch duplex boiler-feed pump which has had no lubrication except

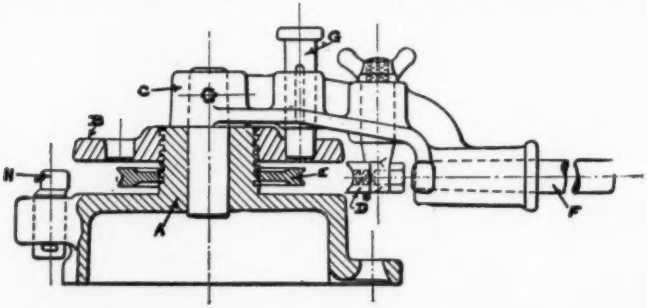
graphite for the past three months, and I have never seen a smoother or quieter working pump, although it is taking the returns direct from a heating system. This pump has been working constantly day and night and every day in the week on about three teaspoonfuls of graphite per twenty-four hours. In the three months we have used in the pump less than two pounds of graphite and hardly a gallon of cylinder oil. It was formerly nothing uncommon to feed a quart of cylinder oil to such a pump every twenty-four hours. This would be 7½ gallons per month, which at 50 cents per gallon, is $3.75 per month, or $11.25 for three months, a matter of $10.35 saved on lubrication in that time. If this amount can be saved on one small pump, what about a plant where there are a number of pumps of various sizes? This same scheme can be used on engines, also, although I should not advise discontinuing the cylinder oil altogether. Aside from economy, this should interest engineers who are using condensed exhaust for feeding boilers.

One reason why some make a failure of graphite as a lubricant is because they use too much, both in cylinders and elsewhere. During my early experience with it the outboard bearing on a 16½ × 48-inch Corliss engine heated up one day, and, as oil failed to produce the desired result, I gave it a bountiful supply of dry flake graphite, and the heating rapidly decreased. I then flushed the bearing with oil and soon had it in normal condition.—"Power and the Engineer."

THE BENDING OF TUBES AND SECTIONS.

At several exhibitions lately considerable interest has been evinced by practical men in the working of a comparatively new machine for bending cold metal tubes and sections. The machine is used by various Government departments, including the British Admiralty, and is doubtless familiar to many of our readers. It is known as the "Kennedy" bending machine, and apart from its workshop application is worthy of attention

as an interesting example of applied mechanics. The application of the machine is made clear from Fig. 1. A is the body, B the adjustable top plate fitting the screw part of the body A. This plate prevents the material bulging and thickening when being bent round the former E. The malleable iron bracket C carries the bending block D and is moved round the machine by means of the lever F. The stop H holds the work against the former E, while being bent. The tendency of the work to expand or thicken when being bent causes the top plate B to bind on the screwed part of the body, and it is necessary to

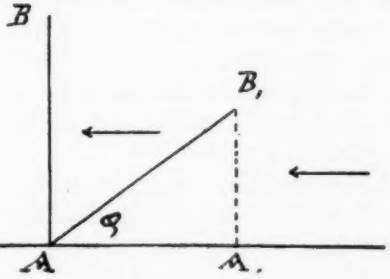


provide means for releasing it when the work is finished; the plunger G fits into any of the four holes shown on the top plate, which is then unscrewed by means of the bending lever F. In the larger machines the roller, in place of the bending block D, is mounted upon a worm wheel and revolved round by means of a worm and pilot wheel, instead of being fixed to the arm C, and moved by the lever F. A short lever is provided to fit the groove in top plate, and the roller stud is brought in contact with short lever when reversing.

When it is desired to bend ordinary iron tubes, a former is first selected corresponding with the required radius, and slightly thinner than the diameter of the tube. This is put in position on the screwed mandrel, the tube to be bent being placed between the former and the stop, and the top plate screwed down by hand until it grips the tube. The bending block or roller is forced round (by means of the lever or worm

wheel as the case may be) until the tube is bent to the angle required. When bending copper or brass tubes of light section, it is of course necessary to have a former and bending block or roller grooved exactly to suit the diameter of the tube.

The bending of tubes with an open seam for beadings and the bending of flat strip iron, tee and channel iron is performed with the aid of special formers, and angle iron, though by far the most difficult of all to bend to a small radius, is readily done on these machines. When bending with the horizontal



web inside, the formers used consist of plain discs fitted with adjusting screws, which are adjusted as follows:

The web of the angle iron is placed under the former (which is held raised from the plate by means of small screws), see Fig. 3; the three grub screws are then adjusted so that the top plate when screwed down, beds on them and just clears the top of the vertical web of the angle iron. The top plate should be screwed down rather tighter than when bending tubes, also it is necessary to have an intermediate bar C of stout material, about 15 inches long, between the angle iron and the bending roller D, to prevent the roller bending the angle at the point of contact. The top plate should be at least three inches larger than the former in order to prevent the angle iron from twisting when being bent. The bracket carrying the bending roller should be adjustable, or else the size of the roller should be such that the material to be bent fits with intermediate bar between the former and the bending roller.

When bending angle iron with the horizontal web outside, the bending roller with intermediate bar must bear on the outside of the vertical web, and not on the edge of the horizontal web, using the same formers as above (Fig. 2).

When bending complete rings, the operator cuts off the desired length and starts from a peg close to former, and as near the end as possible (Fig. 3); he bends partly round the former, then releases the work and holds by clamping screw a few inches from the end (Fig. 4); so that on resuming the bending, the roller with intermediate bar can press over the two ends when completing the circle (Fig. 5). The ends are splayed slightly before bending.

The larger machines can be arranged to drive either by hand or belt power. Being self-contained, they can be readily moved from place to place.

A series of these machines is made by the Kennedy Bending Machine Co. The smallest bends copper and brass tubing from ¼ inch to ⅞ inch outside diameter, 18 gauge and thicker, and standard iron tubing from ⅛ inch to ½ inch internal diameter, with radii from 1⅛ inch to 2½ inches. The largest machine is employed with bending copper and brass tubing up to 2 inches external diameter, and standard iron tubing up to 2 inches internal diameter, with radii from 3½ inches to 7 inches.—"Page's Weekly."

SIMPLIFIED RESULTS FROM FLUE-GAS ANALYSES.

By Frank Kingsley.

Analyses of flue gas have become generally recognized as an indispensable part of a boiler test. They determine not only the completeness of combustion by showing the percentage of carbon monoxide, but also the amount of air used in the combustion of coal, a factor of vital importance in the economical operation of a boiler. However, as is common with all determinations dependent upon more or less complicated formulas, the results obtained are somewhat obscure. Notwithstanding

the ease with which samples of flue gas may be obtained and the simplicity of their volumetric analysis, the bare statement of the composition of the flue gas from a boiler test conveys only a vague meaning.

The methods of determining from the flue-gas analysis the amount of air used per pound of coal while preferable to the direct measurement by manometer of the air entering the ash pit, is at best only a close approximation. Consequently the results may be equally well expressed by means of curves, which show the relations without necessity for involved calculations and have the advantage of presenting an important, but somewhat neglected, subject at a glance.

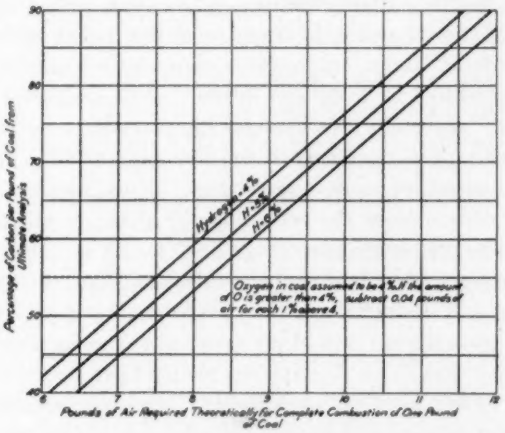


Fig. 1.—Air for Complete Combustion of One Pound of Coal.

The percentage of air used by a boiler furnace, in excess of the actual amount required for complete chemical combination with the fuel, can be obtained from the flue-gas analysis alone, but if the actual weight of air supplied, or gas produced, per pound of coal, is to be obtained, it is necessary to know the chemical composition of the coal in addition to that of the flue gas. Neglecting sulphur, as its presence affects the result less than 1 per cent., and often not at all, the exact chemical re-

quirements for complete combustion of 1 pound of coal may be shown by the curves of Fig. 1.

These curves give only the weights of air required for complete combustion with no excess of air over the chemical requirements, and this, of course, is a condition never realized in practice. Some free oxygen always occurs in flue gas, and without going too deeply into first principles it may be said that, if the coal contained no hydrogen the percentage of excess air supplied to the furnace would be expressed by the relation of the percentages by volume of O and CO_2 as given by the flue-gas analysis. However, as anthracite is rapidly disappearing as a commercial steam fuel, and as all bituminous coals of the United States contain, roughly, 5 per cent. of hydrogen, the above relation does not hold good for the reason that analyses are made at about atmospheric temperature and the water vapor produced by hydrogen in the fuel is condensed and disappears. Naturally this changes the percentage by volume of the remaining components of the flue gas and the amounts of these changes are quite appreciable. With complete combustion of carbon alone the percentage of nitrogen will always be 79 and the maximum percentage of CO_2 will be 21. With the average amount of hydrogen found in coal the nitrogen in the flue gas will be found to run from 80 to about 82, depending mainly upon the amount of air supplied to the furnace; in fact, any flue-gas analysis showing less than 80 per cent. of nitrogen may be looked upon with suspicion. The maximum percentage of carbon dioxide will consequently be only about 18 per cent., the exact percentage depending upon the chemical composition of the coal.

For example, take the case of a representative Ohio coal containing 70 per cent. carbon, 5 per cent. hydrogen and 8 per cent. oxygen. Then if 1 pound of this is burned with exactly the amount of air chemically required the result will be that the carbon unites with $0.70 \times 11.52 = 8.06$ pounds of air, and the hydrogen with $0.04 \times 34.56 = 1.38$ pounds of air.

Assuming 13 cubic feet of air per pound (which value will do as well as any other), these weights of air represent volumes

of 8.06 × 13 = 104.8 cubic feet and 1.38 × 13 = 17.95 cubic feet of air.

Of these volumes 79.09 per cent. is nitrogen and 20.91 per cent. is oxygen; 104.8 × 20.91 per cent. = 21.9 cubic feet O, which forms CO_2; 104.8 × 79.09 per cent. = 82.9 cubic feet N; 17.95 × 20.91 = 3.75 cubic feet O, which forms H_2O, and 17.95 × 79.09 per cent. = 14.20 cubic feet N.

The 21.9 cubic feet of oxygen which unites with carbon will form 21.9 cubic feet of CO_2 and the 3.75 cubic feet of oxygen which unites with hydrogen will form 3.75 × 2 = 7.5 cubic feet of water vapor, in accordance with the law of volumetric chemical combination. The nitrogen will remain inactive and consequently the flue gas from this pound of coal when burned with no excess air will be composed as follows:

CO_2 =	21.9 cubic feet =	17.3 per cent.
H_2O =	7.5 cubic feet =	5.9 per cent.
N = 82.9 + 14.2 =	97.1 cubic feet =	76.8 per cent.
Total,	126.5 cubic feet =	100.0 per cent.

However, as the H_2O will condense and disappear at the temperature at which the analysis is made, the apparent composition will be:

CO_2 =	21.9 cubic feet =	18.4 per cent.
N =	97.1 cubic feet =	81.6 per cent.
Total,	119.0 cubic feet =	100.0 per cent.

If 2½ times the theoretical supply of air is admitted to the furnace and there is 150 per cent. excess air in the flue gas, the composition will be:

CO_2 =	21.9 cubic feet =	7.2 per cent.
N = 97.1 + 145.6 =	242.7 cubic feet =	80.1 per cent.
O =	38.5 cubic feet =	12.7 per cent.
Total,	303.1 cubic feet =	100.0 per cent.

On account of the relatively large variations in the percentages of CO_2 with different coals under the same conditions of air supply, it is impossible to use its relation to the percentage of free oxygen as anything but a rough indication of the percentage of excess air admitted to the furnace. However, as oxygen from the atmosphere is invariably accompanied by 3.78 times its volume of nitrogen, the percentage of nitrogen can be used in place of the percentage of carbon dioxide to establish the percentage of excess air. Upon this fact is based the formula:

$$\frac{\text{Actual air supply}}{\text{Theoretical air supply}} = \frac{N}{N - 3.78 \times O}.$$

Expressed in another form the formula becomes:

$$\text{Percentage of excess air over theoretical requirements} = \frac{O}{(N/3.78) - O},$$

where O is the percentage of oxygen and N is the percentage of nitrogen in the flue gas. N/3.78 is the equivalent of the total oxygen supplied, because the oxygen was accompanied by 3.78 times its volume of nitrogen; and for complete combustion the theoretical requirements are equivalent to total supply less the excess supply, the latter being represented by O. This formula has the advantage that any error in the percentage of nitrogen is divided by 3.78 and become inappreciable in the final result. It may, therefore, be used with accuracy in the form of a curve expressing a direct relation between the percentage of free oxygen in the flue gas and the percentage of excess air supplied to the furnace, values of N being taken as those produced by average quality of coal. These would vary from 82 with no excess air to 80 with 150 per cent. excess (or 2½ times the theoretical air supply). On this basis the solid curve on Fig. 2 has been plotted.

It is understood that these formulas and the solid or the lowest of the four curves of Fig. 2 apply only to cases where perfect combustion takes place. It is sometimes erroneously

assumed that the formula applies to any case, whether or not combustion is complete. In fact, the author of at least one well-known book on steam-boiler economy, in giving this formula, neglects to qualify it as only correct for cases of complete combustion, and in a foot note indicates that it is correct for all cases.

Owing to the fact that carbon monoxide (the product of incomplete combustion, commonly formed in flue gas) unites with only half as much oxygen, by volume, as does carbon by

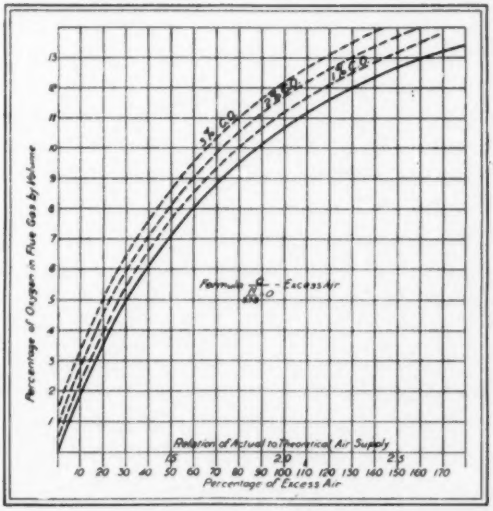

Percentage of Oxygen in Flue Gas by Volume
Formula $\frac{O}{21-O}$ = Excess Air
Relation of Actual to Theoretical Air Supply
Percentage of Excess Air
10 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160 170

Fig. 2.—Relation of Free Oxygen to Air Supply.

dioxide, for any given weights of carbon, it is evident that, when carbon monoxide is present in flue gas, the percentage of free oxygen accompanying it is not a measure of excess air but of unused air, and this prevents accurate use of the formula. A good example of this is shown in analyses of the flue gas from locomotive boilers where the air supply may be actually deficient, and, although not enough oxygen is admitted to burn the coal, enough carbon monoxide is formed (under the peculiar conditions surrounding combustion in a locomotive fire box)

to permit the existence of free oxygen in the flue gas. Under such circumstances the above formula would show an excess of air because of the presence of a small percentage of free oxygen, while as a matter of fact the actual air supply would be less than that theoretically required.

The cases where carbon monoxide is present may be covered by the fact that two volumes of carbon monoxide when burned or combined chemically with one volume of oxygen produce two volumes of carbon dioxide. In other words, 2 per cent.

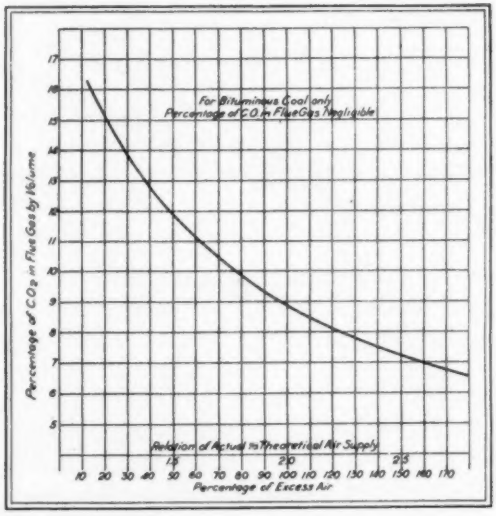


Fig. 3.—Relation of Carbon Dioxide to Excess Air.

of CO in flue gas, if it could have been made to unite with the free oxygen, would have reduced the free oxygen by 1 per cent. of the total volume of gas, without changing the original volume of the CO. Therefore, if a certain coal is being burned with a certain amount of excess air, for each increase of 2 per cent. of CO in the flue gas the amount of free O will increase 1 per cent. This would increase the total volume of the gas so that the real percentage of free O would be 0.99 per cent., but as CO only appears in such small amounts this cor-

rection may be neglected. On this basis then, the dotted curves above the solid curve for complete combustion were plotted, and show that, if, for example, 10 per cent. O with 3 per cent. CO indicates a ratio of actual to theoretical air supply of 1.66, or 66 per cent. excess air, the presence of 10 per cent. O with no CO indicates nearly 90 per cent. excess air.

As mentioned before, the characteristics of carbon dioxide make its quantitative determination easier than those of any of the other components of flue gas, and the expression of the relation of actual air supply to the theoretical supply is very generally given in terms of percentages of CO_2. While this method of expression is not strictly accurate owing to its direct dependence upon the chemical composition of the coal burned in the furnace, the curve in Fig. 3 is given as showing the relation between percentage of CO_2 and the percentage of excess air for such average percentages of nitrogen in flue gas as may be expected for ordinary steam coals. This curve would, of course, apply with sufficient accuracy where the ordinary CO_2 machines are used. In general, for coals with a high percentage of carbon, say 80 per cent., the amount of excess air given would be low by about 5 per cent. For Western coals with only 60 per cent. carbon, the results given would be 5 per cent. too high. These differences apply to average conditions of about 10 per cent. of CO_2, and decrease to practically zero with 7 per cent. of CO_2.—"The Engineering Record."

STEAM CYLINDER LUBRICATION UNDER SUPERHEAT.

By F. H. Corson.

It would appear to be a generally accepted impression that the employment of superheated steam carries with it, as a necessary concomitant, a very serious objection in the form of difficulty of efficient cylinder lubrication, manifesting itself in a greatly reduced life of the working parts. It is with the object of removing or, at least, of considerably modifying this impression that this article is written.

It is perhaps not sufficiently recognized that the use of dry (even if not superheated) steam introduces new conditions into the case, conditions which may and, in fact, will for the most part, necessitate a reconsideration of the principles underlying satisfactory lubrication.

The problem would seem, from the examples frequently encountered in practice, to have received scant consideration, and to have resolved itself into a mere question of delivering a certain quantity of oil into valve chests and cylinders, with no very clear conception of its subsequent behavior. Unfortunately, not only the engines built by lesser known makers but also many carrying the name plates of the finest engine builders are equipped with very inadequate provision in this respect, and rarely does the lubrication appear to have been treated with the care bestowed upon the general design of the engine.

The methods adopted are to a great extent such as to preclude entirely the possibility of successful lubrication, and the fact that many engines have shown good performance under these conditions is an eloquent testimony to the wetness of the steam, and also to the lubricating qualities of a film of water between the working surfaces. When by means of the superheater this water is removed, trouble is at once experienced, and a rapid increase in the rate of wear occurs. It is often considered that, as the arrangements have given good results before the advent of superheat, the source of the difficulty must be sought elsewhere. The oil is changed, and when perhaps little or no alleviation is achieved, the position is accepted and the increased repair bill is regarded as an unavoidable setoff against the advantages derived from superheat.

It is submitted that a more careful investigation into the matter would reveal the inefficiency of the existing devices, and that, in most cases, a more rational design would remove the greater portion if not the whole of the difficulty. Premising the ability of the oil to withstand for a short time the action of steam at a high temperature, there is intrinsically no insuperable difficulty in the way of a satisfactory solution, provided that the question be approached from first principles.

It is well known that few, if any, oils are able to resist unchanged, for long periods, exposure to the influence of high temperature, this treatment resulting in the rapid evaporation of the lighter constituents and the deposition of residues resembling coke, rubber or leather, according to the characteristics of the particular oil used. Such residues are familiar to most engineers, and are readily observed on the parts of the engine, *i.e.*, steam receivers, where the surfaces are not subjected to the scouring action of steam at high velocity.

It is at once seen, then, that the intermittent supply of a comparatively large quantity of lubricant is likely to be productive of positive harm by the formation of carbonized deposits, and, consequently, that the delivery of oil to the engine must be continuous in point of time. Briefly stated, the two principal essentials are this continuity of supply and certainty that the oil is being carried to the working faces.

Too frequently it would appear to be unappreciated that the conditions encountered in actual operation are entirely different from those obtaining when the engine is at rest, and that the supply of lubricant at a point which would be eminently suitable under the latter condition may be totally ineffective under the former. An instance of common occurrence is the fitting of the oil feed to the center of the cylinder cover of an inverted vertical engine. A moment's consideration shows that any oil admitted at this point is, under the most favorable circumstances, swept across the cover by the rush of steam from the port, lubricating one side of the cylinder only, while on the exhaust stroke the oil supplied is carried away immediately with the outgoing steam. Where the cover is recessed to accommodate the end of the piston rod, it is at least probable that the oil trickles down the walls of the recess, undergoing carbonization before arriving at a point whence it can be swept onto the cylinder walls.

To feed oil into the cylinder or valve chest through the cover is, in most cases, wasteful, and involves the direct rejection with the exhaust steam, and without doing any work, of at least half the quantity. The impression, which would seem to

be fairly widespread, that the lubricant enters the cylinder in the form of drops, is seen to be erroneous on the slightest reflection. It must be remembered that at high temperatures and pressures the viscosity of all oils is somewhat less than that of water at ordinary temperature. The impossibility of forming a drop of such a liquid on a surface traversed by steam with a velocity of 100 feet per second, or, for the sake of illustration, in a gale of 60 miles per hour, is obvious, the lubricant being, in fact, carried away as fast as it appears.

The secret of successful cylinder lubrication lies in the thorough atomizing and admixture of the oil with the steam, rather than in any attempt to apply it directly to the working surfaces. It is evident that if oil is entrained in the form of minute particles in the steam itself, it is carried into immediate contact with all surfaces exposed to the latter.

The writer has recently carried out some interesting experiments in which a 900-horsepower six-cylinder compound engine has been lubricated by one feed on the boiler side of the stop valve. The oil pipe was in this case carried to the center of the steam pipe, delivering its contents full into the current of steam, and the quantity of oil used was far smaller than that required by the earlier arrangement. The internal condition of valves and cylinders proved on examination to be excellent, demonstrating very efficient lubrication, and showing a great improvement on the results obtained where each high-pressure cylinder was fed direct. This is perhaps an extreme case, and this method is not applicable indiscriminately to all engines.

The effect of large receivers, introducing reduction in velocity and change in direction of the steam, must receive due consideration. The principle of lubrication of the steam primarily, if such an expression may be permitted, and, through this, of the working parts, is vital, and, this principle borne in mind, its application in individual cases should present no great difficulty.—"Power and The Engineer."

CAST-IRON FITTINGS AND SUPERHEATED STEAM.

By JOHN PRIMROSE.

Articles have been appearing in engineering papers and manufacturers' publications in which superheated steam has been charged with being responsible for the failure of cast-iron fittings and for trouble due to leaky valves in steam-pipe connections. In cases where there has not been time nor opportunity to investigate thoroughly, the result has been rather to militate against the use of superheat, and in many cases people have been afraid to avail themselves of the increased economy and other advantages of superheated steam. Statements to the effect that all cast-iron fittings and valves must be replaced by cast steel, if superheated steam is to be used safely, are most unfortunate at the present stage of development of superheated steam in this country, because the popular knowledge of the subject is such that the effect of such a statement is to picture untold troubles of all kinds with the steam equipment for which no possible increase in economy would be a recompense. It is a pity to block an advance in steam economy of such undoubted value unless, of course, there is some real danger in the way.

In a number of cases, cast-iron fittings through which superheated steam was passing failed in various ways, and the conclusion arrived at was that superheated steam weakened the metal. The complete lack of a satisfactory theory or reason why superheated steam should weaken cast iron and make fittings and valves made of that material dangerous arouses suspicion of the correctness of the conclusion. Test bars were carefully cut from some of these fittings and the breaking strain compared with what it should be for an average run of iron. The metal of the broken fitting showed, in some cases, but half the tensile strength, and the conclusion arrived at was that superheated steam passing through a fitting reduced its tensile strength at least 50 per cent. The logic whereby the conclusion is obtained is not apparent. The fitting was evidently weak, otherwise it would not have failed; so that nothing has been learned by breaking test bars. Further, cast-iron

fittings have been failing in saturated-steam lines since the metal was first used for the purpose, easily accounted for by the well-known difficulty of making homogeneous castings and the difference in the tensile strength of castings from the same heat. It is not remarkable, then, that several cast-iron fittings have failed when passing superheated steam, and the cause is much more likely to be the original weakness of the metal rather than the effect of superheat. Cast-iron fittings carrying a fluid so far removed from superheated steam as water have been known to decrease in weight and fail in different ways. Certain it is that cast-iron fittings have failed in many ways before the advent of superheated steam in this country, and it seems unreasonable that a few subsequent failures should be considered proof that superheated steam is injurious to cast iron.

One manufacturer of cast-steel fittings has made elaborate tests, exposing cast-iron flanges to varying temperatures from 500 to 800 degrees Fahrenheit, for a number of weeks. The effect on a flange 12½ inches in diameter was to increase the diameter $\frac{19}{100}$ of an inch. Upon this is based a theory that a molecular change in the metal has resulted, the increased volume indicating loss in tensile strength, and that possibly the carbon silicon and phosphorus contents of the metal are so affected by the temperature that the strength is impaired. There are very many reasons why this latter is scarcely possible, but assuming for the sake of argument only that cast iron does "grow" and lose strength at these varying temperatures, why condemn superheated steam? If superheated steam were allowed to vary in temperature to any such extent as these tests, the damage to fittings and valves in the steam line would be quite inconsiderable compared with the difficulties in operating the engines or turbines.

Constant temperature is essential to successful operation whether saturated or superheated steam be used, and it is quite possible with properly designed superheaters to keep a temperature within 10 degrees either way of a determined point, or a maximum variation of 20 degrees. It is quite true that in some cases superheaters have been so designed that the temper-

ature fluctuates and troubles have followed, but the troubles with the pipe-line fittings have been the least of these, and the cause has been not the temperature due to superheat, but the fluctuations in temperature due to the design of the superheater, in the same way that a fault in the design of a boiler, engine or condenser is likely to be heard from.

So much has been charged against superheated steam that an investigation was started to find out what the people of most experience in the use of superheat in this country and abroad had found out concerning the effect of superheated steam on cast iron.

In Europe superheaters are always a part of a power plant of any importance, and superheated steam has been in common use there for more than thirty years. If, in the few years during which superheated steam has been used in this country such destructive characteristics have developed, the Germans with their thirty years of experience ought to be in a position to tell us about it. A well-known German engineer was recently in this country, and the matter was brought to his attention. He was interested and surprised that he had not learned before of the effect of superheated steam on cast iron and promised to look the matter up on his return home. He has since written that he could not find anything in their engineering literature to bear out this contention, or anyone who believed that such a thing was possible. This engineer's experience with superheated steam extends over twenty-five years or more, and shows that highly superheated steam—600 degrees Fahrenheit—has no effect on cast iron. He uses fittings of gray-iron castings, except where the Government specifications insist upon cast steel. The Government insists upon cast steel at times, but with regard to the pressure of the steam rather than to the temperature, and is not influenced by the steam being saturated or superheated.

Investigations in England, where superheaters have been used for a much longer time than in this country, have resulted in much the same information. A member of the engineering staff of one of the steam-users' associations, corresponding to

our insurance and inspection companies, writes that he "has not found any reduction in strength of either cast iron or steel due to temperatures obtained in practice." As in Germany, cast steel is used for fittings above certain pressures, but it is because of the pressure and not at all because the steam is superheated. Different steam-users' associations in England have been consulted, and they have all agreed that superheated steam does not weaken cast iron in their experience.

In this country there exist many proofs of the fallacy of the theory that superheated steam weakens cast iron. A number of superheaters built entirely of cast-iron pipes and headers were installed in 1901—eight years ago. These superheaters are located in the settings of Babcock & Wilcox boilers above the first pass of tubes, directly in the path of and surrounded by gases at 1,000 to 1,200 degrees Fahrenheit. These superheater tubes are, of course, cooled by the circulation of the steam at 175 pounds pressure and superheated 150 degrees, but must be considerably hotter than the temperature of the steam, or than the usual temperature of cast-iron fittings, in steam lines distributing the superheated steam. These superheaters are still in successful operation and have cost nothing for repairs. About two years ago it was necessary to move one of the units containing one of the superheaters. The superheater had to be taken apart and was carefully examined. Absolutely no evidence existed of any deterioration in any way, and the superheater was reërected in the boiler and went together as easily as when first installed. The superheater tubes are $7\frac{1}{2}$ inches inside diameter, 12 feet 6 inches long, made of good gray iron, care being taken to secure sound castings. The people who built this superheater have changed the design of the superheater tube somewhat, although retaining cast iron as the best material to meet the hot gases, and have installed upward of 1,000,000 horsepower of superheaters, and, except in a very few isolated cases, have used cast iron for all fittings connecting the superheater to the boiler and its various sections to a common outlet. Some of these fittings are in contact with hot gases, being inside the boiler settings, and have been in service since 1901.

In not one single instance has trouble been reported in any of these fittings which could be traced in any way to the effect of superheated steam, and this company still continues to use cast iron for all fittings and considers it to be the best material for the purpose.

The result of investigation at a plant where a large cast-iron fitting showed signs of weakness, and superheated steam was charged with being to blame, is interesting in this connection. The plant consists of thirty-two water-tube boilers arranged in two decks, sixteen on the upper floor and sixteen below. A steam header connects the boilers on each side of the upper boiler room and these headers come together in a tee at the end of the boiler room. This tee has an opening looking down which connects with a larger tee below, taking the steam from each side of the boiler room on the lower floor. The main steam line to the engines carrying all the steam generated starts from this tee. The large tee taking steam from the upper and lower boiler-room floors was the tee that failed. It showed surface cracks and was distorted. Investigation disclosed that only fourteen of the boilers on the upper floor were equipped with superheaters, and that these superheaters were good for but 75 degrees of superheat. The superheated steam from these fourteen boilers was mixed with the steam from eighteen boilers furnishing saturated steam containing the usual percentage of moisture, so that the steam at the tee which failed could not have been superheated more than 30 degrees, and even the most ardent critics of superheat insist that more superheat than this is necessary to bear out their arguments. Further investigation disclosed other interesting facts. All of the fittings on the boiler and superheater outlets were of cast iron. The fittings on the boiler outlets downstairs (saturated steam) were all affected in the same manner as the large tee just described. The fittings on the superheater outlets upstairs (superheated steam) showed no such effect. The explanation is evident. The large tee most seriously damaged was subjected to varying temperatures, and consequently continually changing strains, due to the mixture therein of superheated steam with

steam containing moisture. The fittings passing saturated steam only were probably attacked by some impurity in the feed water, while the fittings passing superheated steam escaped because all of the moisture was evaporated in the superheater and in this way were protected against the injurious action of the saturated steam. Here is a complete reversal of the situation, and yet this particular case has influenced many persons against superheat.

In addition to all this evidence exonerating superheated steam, and in order to be sure that at temperatures proper for use in engines and turbines it could have no possible effect on cast iron, a prominent foundryman, himself engaged in manufacturing steel fittings, and a well known metallurgist were consulted. Samples cut from cast-iron fittings on the inlet (saturated) and outlet (superheated) connections of a superheater in use for more than five years and superheating steam to 550 degrees Fahrenheit, were photographed under a miscroscope and carefully examined. The opinion of these experts is that there is no reason to expect any graphitic change in the iron at temperatures less than 700 degrees Fahrenheit. The only graphitic change which would seem possible in the metal would be the changing of graphitic to combined carbon, which would result in the hardening of the metal, slightly increasing its tensile strength. It is customary in annealing furnaces to use a temperature in excess of 900 degrees Fahrenheit to produce any effect in gray iron. Micro-photographs taken of samples from the center of the cross-section of the inlet and outlet fittings showed no more difference in the amount of carbon present than would be expected at different points in the same cross-section, proving that there was no change in the carbon conditions by reason of superheated steam. Photographs of the edge of the polished cross-section were also examined, showing the close-grained iron which always occurs near the outlet surface, due to chilling contact with the sand in the mold. Here again the experts report no change in the condition of the carbon by reason of exposure to superheated steam. Photographs were also taken of the polished surfaces

after being etched with acid of different samples from inlet and outlet fittings. but examination showed no effect of superheated steam.

The foregoing statements evidently prove that something more than superheat is necessary to destroy cast-iron fittings. Unusually high temperatures (it is possible to convey steam superheated as high as 1,200 degrees Fahrenheit, and it is being done in this country) undoubtedly need special treatment, but for the ordinary steam-power plant the best results are obtained by superheating the steam to a final temperature not exceeding 500 degrees and in designing the superheater to maintain a closely constant temperature. At this temperature and under these conditions no ill effects need be feared from cast-iron fittings, nor any other parts of the equipment. Sudden variation in temperature, or changes from superheated steam to steam containing large quantities of moisture are bound to result in troubles of all kinds, but the design of the superheat is to blame.—"Power and The Engineer."

ROTREX HIGH-VACUUM AIR PUMP.

The steadily increasing demand for higher economy of steam prime movers which has been accompanying the development and application of steam turbines during the past few years made a simultaneous advance in the design of condensing apparatus necessary. Guided by such considerations, the C. H. Wheeler Manufacturing Company has developed a new type of rotary air pump embodying the essential features of a high-vacuum air pump occupying a comparatively small space, with the advantage of being adapted for direct connection to another auxiliary.

Fig. 1 illustrates one of the standard arrangements, consisting of a vertical steam engine direct connected to a Rotrex air pump on one side and a centrifugal circulating pump on the other. By this arrangement one engine drives both pumps, the speed of the combined outfit being from 180 to 250 revolutions

per minute, according to the size of the unit. This eliminates the additional steam cylinders ordinarily used to drive the air pump.

The construction of the Rotrex pump is shown in Fig. 3. It consists of a light-weight cylindrical casing, one rotor eccentrically mounted on a heavy steel shaft carried in outboard ring-oiled bearings, independent of the stuffing boxes. Division between the suction and discharge in the pump cylinder is made by means of a radius cam, which is carried in independent bearings and operated by means of a lever and crank from the rotor

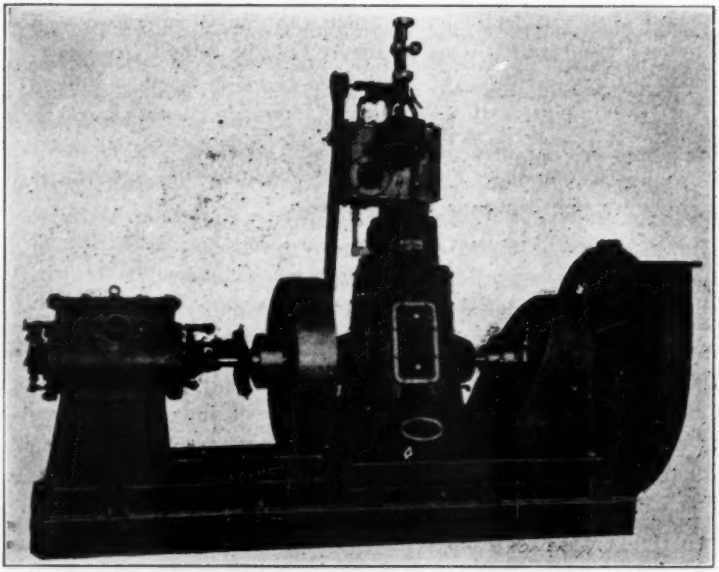


Fig. 1.—ROTREX AIR PUMP AND CENTRIFUGAL PUMP DIRECT-CONNECTED TO A VERTICAL STEAM ENGINE.

shaft on the outside of the pump. By this arrangement internal contact is eliminated; the rotor operating with a close clearance from the bore of the pump cylinder and the cam maintaining a close clearance from the rotor. The clearances, by an ingenious arrangement of ports, are thoroughly water

sealed at all times, *i. e.*, the pump relies on water-sealed clearances to produce the high vacuum obtained, and not surface contact nor sliding fits. The other notable feature of this pump is a special arrangement of metallic discharge valves, no suction valves being used, thus practically eliminating expensive upkeep.

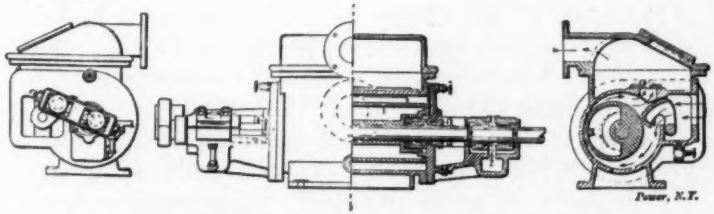


Fig. 3.—Showing Construction of the Rotrex Pump.

The pump is being made in capacities from 50 to 5,000 horsepower in one unit; beyond that size multiple units are used, an independent air-pump engine having one Rotrex pump direct connected to each end of its crank shaft.—"Power and The Engineer."

NAVAL NOTES.

UNITED STATES.

The Trial of the "North Dakota."—The following are some of the general results of the trial of the U. S. S. *North Dakota.* It is contemplated to publish a complete report in the next number of the JOURNAL.

Average of five high runs on standardization trial: Speed, 21.833; S.H.P. of main turbines, 33,875; revolutions, 286.22.

Four hours full speed (results for three hours.)—Speed, 21.64; S.H.P. of main turbines, 31,300; revolutions, 280.42.

At eighteen minutes after beginning of fourth hour of full-speed run a boiler tube gave way. Four boilers were cut out and the speed for the last hour dropped to such an extent that the average speed for four hours was 21.005 knots.

Twenty-four hours at 19 knots.—Speed, 19.245; S.H.P. of main turbines, 16,755; revolutions, 231.955.

Twenty-four hours at 12 knots.—Speed, 12.066; S.H.P. of main turbines, 3,800; revolutions, 143.16.

The following water-consumption results were determined:

Speed	12	19	21
Water per S.H.P. per hour, main turbines only, pounds	20.6	14.25	13.8
Speed, knots	12.1	19.245	21.64
Water per S.H.P. per hour, main turbines and engineer auxiliaries, pounds	23.94	15.92	14.408
Speed, knots	12.066	19.245	21.64
Total water per hour, main turbines and engineer auxiliaries, pounds	90,984	266,711	450,965
Speed	12	19	21
Total water per hour for main turbines only, lbs.	76,000	238,000	349,000

ENGLAND.

Two New British Warships.—Two more ships for the British Navy have just been launched, the cruiser-battleship *Indefatigable* at Devonport, and the cruiser *Gloucester* on the

Clyde. The designs for the former represent the latest devlopment of the *Invincible* idea, and although before she enters her first commission still more powerful vessels will be on the stocks, she will for a time be the most powerful cruiser in the world. The *Invincible* and her sisters, the first cruisers to have an all-big-gun armament, are, it will be recalled, 560 feet long, with a beam of 78½ feet, and a displacement of 17,250 tons. The *Indefatigable,* which for the moment stands alone, is 20 feet longer, 1 foot broader, and displaces 18,000 tons. She is designed also to exceed the speed of the three sisters by one knot, as she is expected to attain 26 knots, and with that object in view her turbines will develop 45,000 horsepower. Her armament and its arrangement will be similar to that of the *Invincible,* but she will carry twenty 4-inch guns in place of sixteen, the main guns being 12-inch fifty-caliber pieces paired in barbettes with heavy shields, as before.

What is to be the next step in the progress of British cruisers is not generally known, the Admiralty having preserved the secret well so far, but it is said that the next vessel will be 600 feet long, will displace 25,000 tons, will be propelled at 30 knots by turbines of 70,000 horsepower, and will have eight 13½-inch guns instead of eight 12-inch. All these statements require to be received with caution, particularly the assertion that she will carry 13½-inch guns. It is not improbable that further advances will be made with 12-inch guns before the very serious step of changing the standard armament is adopted. Moreover, special forgings and an entirely new design of mounting would be required, and as far as we have been able to ascertain, the principal makers have not yet been approached on either point, save in a very tentative way. It has always been recognized that a bigger gun might some day be necessary, and gun-makers have prepared designs, whilst a 13½ gun has been made at Woolwich, and is now on board the *Excellent.* There is, therefore, some grounds for the rumor, but it still lacks confirmation.

The *Gloucester,* launched by William Beardmore & Co., is a second-class cruiser, one of five, all being built by private

firms. She is 453 feet long, and displaces something under 5,000 tons. Turbines indicating 22,000 horsepower will propel her at 25 knots. It is reported that the same builders have just received the order to construct another vessel of the same type improved.—"The Engineer."

The New British Dreadnoughts and Inflexibles.—Rather complete particulars have lately been made public of the latest British *Dreadnoughts* and *Inflexibles,* which are now being built in Government and private dockyards. Taking the *Neptune* as the latest representative of the *Dreadnought* type in the British Navy, we find that the length has been increased by 20 feet and the beam by 4 feet, and that the displacement has been increased from 17,900 tons to 20,000 tons. The speed, 21 knots, remains the same, and no changes of any consequence have been made in the disposition of the armor for the protection of the hull or the barbettes and turrets.

The most important changes—those which serve greatly to increase the power of this ship as compared with the original *Dreadnought*—relate to the armament. In the *Dreadnought,* it will be remembered, ten 12-inch guns were mounted in the following positions: Two on the forecastle deck; a pair on each beam amidships on the main deck, with the superstructure between them; and four in two turrets on the main deck astern and on the center line of the ship. This plan has been changed in the *Neptune* by placing the two wing turrets *en echelon,* or diagonally, with sufficient distance between them in the fore-and-aft direction to permit the guns of both turrets to fire on the same broadside. Another change is to raise these two turrets and also turret number 4, one deck higher, placing them at the same elevation as the forward turret. The aftermost turret will be located, as in the *Dreadnought,* on the main deck. By this redistribution the *Neptune* can fire six guns ahead, eight astern, and ten on each broadside, as against six ahead and astern, and eight on each broadside, in the original *Dreadnought.* The *Neptune* will carry a new 50-caliber, wire-wound 12-inch gun, and not, as reported, a 13.5-inch gun. For torpedo attack a battery of 4.7-inch guns will be mounted in a

lofty central armored redoubt surrounding the smokestacks, which will protect both the guns and the smokestack bases.

The improved *Inflexible,* known as the *Indefatigable,* has 25 feet more length, 2 feet more beam, and 2,000 tons additional displacement than Admiral Seymour's flagship. The speed is the same; but the extra 20 feet of length will enable the mid-ship turrets to be placed farther apart in the fore-and-aft direction than they are in the *Inflexible,* with the result that the broadside angle of fire of what might be called the "off turret," that is to say, the turret which is on the side of the ship remote from that on which an engagement is taking place, will be greatly increased.

Great Britain evidently is well pleased with her 26-knot battleship cruisers of the *Inflexible* type, for she is now preparing to lay down on the ways vacated by the *Indefatigable* another ship of the same type but of far greater dimensions. She is to be 600 feet in length, and equipped with turbine machinery of even greater horsepower than that installed in the *Lusitania* and *Mauretania.* As these liners exert over 70,000 horsepower when they are making their maximum speed of 26 knots, it can be understood that to secure the 28 knots required in the new cruiser-battleship, the horsepower must run up to 80,000 or more. It is probable that the vessel, on her trials, like her predecessors of the *Inflexible* class, will exceed the requirements by about a couple of knots. The *Inflexible* and her sisters made 28 knots over short distances, and it is likely that the 600-foot ship will be able to carry her battery of eight 12-inch guns for a short spurt across the high seas at a speed of 30 knots an hour, which is higher than the average speed of the torpedo-boat destroyers.—"Scientific American."

Launches.—The new first-class battleship *Neptune,* one of the improved *Dreadnought* class, was launched on Sept. 30th, at Portsmouth, the naming and launching ceremony being performed by H.R.H. the Duchess of Albany. The dimensions of the ship are understood to be as follows: Length 510 feet between perpendiculars, 560 feet over all; beam, 85 feet, and displacement, 20,250 tons. She will be fitted with turbine engines, to develop 25,000 I.H.P., giving a speed of 21 knots.

Her armament will consist of ten 50-caliber 12-inch guns of the latest pattern, disposed in five turrets, three of which will be on the central line, and one on each side of the ship, these latter being placed *en échelon*. She will thus be able to fire the whole of the ten guns on either broadside, whereas her predecessors could only fire eight; in addition, the middle turret on the center line will be raised and placed close forward of the after one, so that its guns can fire over it, a stern fire of eight guns, instead of six only, will be thus obtained. The secondary armament will, it is believed, be the same as that of the *St. Vincent*, viz: twenty 4-inch guns, but in some quarters it is stated that she will carry 6-inch guns; there will also be three submerged torpedo tubes.

On the same day the new second-class cruiser *Glasgow* was launched from the works of the Fairfield Shipbuilding and Engineering Co. at Glasgow. She is the first of the five new cruisers of the "Town" class to take the water, the contracts for which were placed in November last at Fairfield, Clydebank, Dalmuir, Barrow and Elswick. Her dimensions are as follows: Length, 453 feet over all; beam, 47 feet; draught, 15 feet 3 inches, with a displacement of 4,820 tons. The ship will be provided with Parsons turbines to develop 22,000 I.H.P., and give a speed of 25 knots. The armament will consist of two 6-inch Q.F. and ten 4-inch Q.F. guns, with two 18-inch broadside submerged torpedo tubes; one of the 6-inch guns will be mounted on the forecastle, where the conning tower—constructed of special steel—will also be placed, the other 6-inch gun will be mounted aft, and five of the 4-inch guns on each broadside. Protection to the vital parts will be provided by a curved-steel protective deck extending the whole length of the ship, and further protection to the machinery will be afforded by the bunkers.

FRANCE.

Report of the Parliamentary Commission of Enquiry into the State of the Navy.—The following abridgement of the *Rapport* of the Parliamentary Commission, under the presi-

dency of M. Delcassé, appointed by the Chamber to report on the condition of the Navy, is given by the *Temps:*

I.—*Naval Construction.*

The Commission having ascertained that in the last ten years Parliament has been asked to authorize the commencement of ships, the plans of which, for the most part, had not been definitely decided on;

That months and years generally separate the different contracts for hull, turrets, boilers, etc., entailing a considerable loss of time and money, and that this has been notably the case with the battleships of the *Patrie* class and the armored cruisers *Waldeck-Rousseau, Ernest-Renan* and *Edgar-Quinet;*

That, moreover, important changes have been made during the building of ships, notably as regards the artillery of the *Justice, Démocratie, Liberté, Vérité* and the armored cruisers *Ernest-Renan, Waldeck-Rousseau* and *Edgar-Quinet,* changes which tend to destroy homogeneity—one of the most important characteristics of a good fleet;

That the greater part of these defects are aggravated in the six battleships of the *Danton* class, the preliminary contract for which, signed in December, 1906, has had hundreds of changes made in it since;

Proposes to the Chamber:

1. To severely blame such proceedings, which are prejudicial to the public finances, and incompatible with rational, methodical and rapid construction.

2. And that it should decide that henceforth it will only authorize the construction of a ship after receiving assurances that the plans are at least in all essential particulars definitely drawn and the contract ready for signing.

II.—*Arsenals.*

The Commission having ascertained that our naval arsenals are not actually in a condition to satisfactorily and with requisite rapidity carry on at the same time both new construction and repairs to ships of the fleet;

That tools and machinery are, generally speaking, insufficient and out of date;

That the abolition of piecework coincident with a reduction in the daily hours of labor, and the weakening of the power and authority of the heads of departments, has been the consequence of a notable diminution in production;

That a want of material has sometimes caused a stoppage of work;

Proposes to the Chamber to decide:

That in the interests of the Navy, as well as of the public finances, our arsenals be supplied with modern tools and machinery to admit of a *maximum* production in a *minimum* time.

That it must be firmly maintained that work goes on for a full eight hours daily.

That piecework be re-introduced as far as possible, together with equitable remuneration, experiences at Brest and Ruelle having shown that its introduction has produced excellent results.

That in certain workshops, where it would not be advisable to introduce piecework, the workmen to receive sufficient wages to make up the difference.

That the decree of 13th June, 1907, be revised so as to give heads of departments greater powers of reward and punishment.

That before sending any work to be carried out by a private firm it be ascertained whether the arsenals are in a position to undertake it.

And for the prompt execution of work often delayed by reference to the Central Authority, greater power of decision be vested in the *Préfets Maritimes* and the Commanders-in-Chief of the squadrons.

III.—*Contracts for the Navy.*

The Commission having ascertained:

That prices of materials for the Navy have increased of late;

That the prices of turbine engines in particular appear excessive;

That this increase in price may be attributed in great measure to the fact that the Navy Contracts department does not stand out sufficiently against the demands of contractors, and that too many useless clauses are inserted in the contracts;

Proposes to the Chamber:

To request the Control department to endeavor by every means to obtain a reduction of prices, particularly by the suppression of certain penalties in the terms of contract which are of no use to the Navy.

To abolish, except in certain cases, all premiums to contractors, simply requiring the results which are thought necessary and sufficient.

IV.—*Boilers.*

The Commission proposes that the Chamber should express its regret:

1. That for the past ten years Ministers of Marine have neglected to employ competition to obtain a reduction of prices in the purchase of two boilers which are considered the best;

2. That the Minister of Marine, having in a letter of 5th August, 1905, consulted the technical committee as to the type of boiler to be adopted for the *Edgar-Quinet* and *Waldeck-Rousseau,* did not afterwards apply for tenders from the four firms designated by the committee, but selected two firms only from the list supplied;

3. That for the six battleships of the *Danton* class, the Minister, without even consulting the Technical Committee, took upon himself to order an equal number of boilers from the before-mentioned two firms;

4. That in his selection of boilers the Minister has not, in the majority of cases, chosen the type which the Commission of Enquiry have found to be most in favor with naval engineers and mechanicians;

And decides that henceforth, unless it be explained to Parliament that it is impossible, tenders shall always be asked for from all the firms recommended by the Technical Committee.

V.—*Artillery.*

The Commission having ascertained:

That the four divisions of battleships and the division of armored cruisers of the Mediterranean Fleet have not their full allowance of steel shell on board;

That the two divisions of armored cruisers of the Northern Squadron have not the third of their regulation supply of those projectiles;

That for both the Mediterranean and Northern Squadrons there is no reserve stock of steel shell ready;

Considering:

That it appears from the statements of the Minister of Marine and the artillery department that the problem regarding steel explosive shell was decided for small-caliber guns in 1905 and for heavy guns in 1906:

Proposes to the Chamber:

To express its serious regret that the necessary diligence has not been shown during the last three years to furnish our squadrons with this projectile and to constitute a reserve stock.

The Commission having also ascertained:

That it was only towards the end of February, 1909, that the pattern of shell was decided on for the six battleships of the *Danton* class which were ordered in December, 1906, and are due to be completed in December, 1910, and that the time is now too short to allow of the necessary quantity of shell being manufactured by the latter date, and that also no credit has been taken in the Budget of 1909, for this purpose;

Proposes to the Chamber:

To blame the improvidence and dilatoriness which this situation reveals.

VI.—*Docks and Arsenals.*

The Commission having ascertained:

That the program of works drawn up in 1901 to place the military ports in a position to deal with the new ship construction of 1900 has only been partly executed;

That nothing has been done up to the present to provide

docking accommodation for the battleships of the 1906 program:

Proposes to the Chamber:

To blame the improvidence and carelessness which these facts display.

VII.—*Central Administration.*

The Commission having ascertained:

That neither unity of views, coöperation, method nor definite responsibility exists between the different branches of the central administration, and that there is too often negligence and confusion;

Proposes to the Chamber:

To order a reorganization of the central administration, which will assure co-ordination of effort and leave the head of each great department responsible for carrying out the measures which have been decided upon after collective deliberation under the presidency of the Minister.

This report of the Commission of Inquiry has been, and still is being debated in the Chamber, and resulted indirectly in the fall of the Clemenceau Ministry on 18th July.

Committee's Report on the "Danton's" Mishap.—The committee ordered to inquire as to the causes of the failure to launch the battleship *Danton* at Brest has reported the following reasons:

1. The new method of greasing the ways resulted in a part of the fatty matter being absorbed and part floating away on the water, which rose higher than expected on the day of launching.

2. The resistance presented by the large area of the stern when taking the water.

3. The incline given to the ways was not sufficiently steep.

4. The *Danton* having started under bad conditions, moved slower and slower until she brought up.

The Naval Program.—It is stated that the *Conseil Supérieur de la Marine* at its recent meeting at the Elysée came to a definite conclusion regarding the future constitution of the French fleet, which is to be as follows:

45 battleships; 12 scouting cruisers; 60 destroyers (sea-going); 84 destroyers for mobile defence; 64 submarines.

It will be remembered that the 1900 program is as follows:

28 first-class battleships; 24 armored cruisers; 52 destroyers; 263 torpedo boats; 38 submarines.

And when the Budget for 1907 was introduced with the *Danton* class of battleship it was stated by the Admiralty that in theory the constitution of the fleet had been fixed as follows:

38 battleships; 20 armored cruisers; 6 scouts; 109 destroyers; 170 torpedo boats; 82 submarines (for offence); 49 submarines (for defence).

Taken together, these three programs show that the constitution of the fleet is to be changed both in the number of armored ships and the importance assigned to each class of ship. The number of battleships has increased, armored criusers have disappeared, as also have torpedo boats. Less importance is given to submarines, and in their place scout cruisers and destroyers are given greater prominence. Thus, according to the program of the *Conseil Supérieur de la Marine,* real military value is only given to the battleship, the scout cruiser and the destroyer.

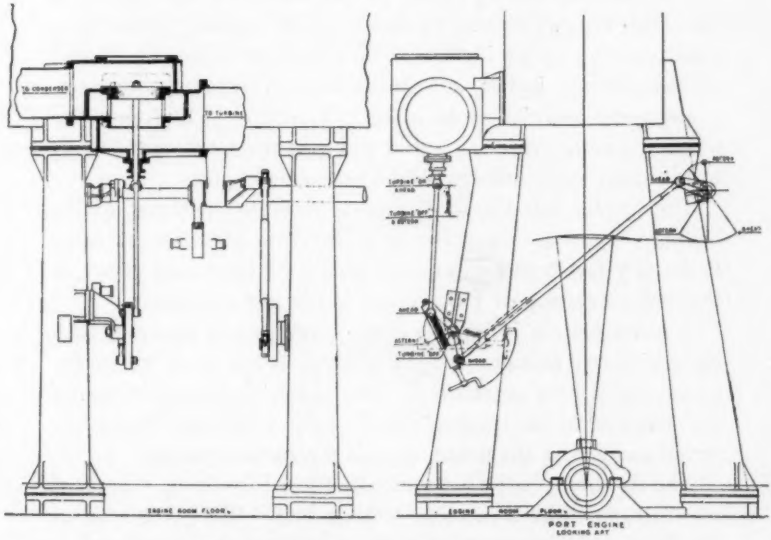
The date limit for the completion of the battleships is given as 1925, and as the life of a battleship is fixed at 25 years, it follows that at that date the only battleships now built fit for service will be the six of the *Patrie* and six of the *Danton* class. In the sixteen years to come, therefore, it will be necessary to build 33 new battleships, 12 new scout cruisers and more than 100 destroyers, besides submarines, so that this new program is a vast and expensive one.—"J. R. U. S. I."

TRIAL PERFORMANCES OF S.S. "OTAKI."*

The engines of the *Otaki* consist of two sets of ordinary triple-expansion reciprocating engines, driving wing propellers, and a low-pressure turbine driving a center propeller. The arrangement is shown in Figs. 1, 2 and 3.

* Extracts from paper by Engineer Commander W. McK. Wisnom, R. N. Read at the joint meeting of the Institution of Engineers and Shipbuilders in Scotland and the North East Coast Institution of Engineers and Shipbuilders.

In ordinary ahead working the reciprocating engines exhaust into the turbine, which can only revolve in one direction; but change valves are fitted, so that the reciprocating engines can also exhaust direct to the condenser. These change valves are operated by gear from the reversing shafts; the act of reversing the reciprocating engines closes the connections between them and the turbine and opens direct communication from the reciprocating engines to the condensers. The turbine



Figs. 4 and 5.—Change Valve Gear.

is thus automatically thrown out of action when going astern. The operating gear for the change valves is also arranged so that the turbine can be cut out at any time without stopping, and the ship can then be maneuvered as an ordinary twin-screw vessel. The arrangement also permits the turbine to be used in combination with one reciprocating engine only for ahead working. The arrangement of change valves and gear is shown in Figs. 4 and 5.

Two condensers are fitted, and each set of reciprocating

engines exhausts direct to its own condenser when the turbine is not in use. The turbine exhausts to both condensers. A shut-off valve is fitted in the exhaust from the turbine to each condenser, so that in the event of a defect in either condenser, or its pumps, or other connections, the corresponding shut-off valve can be closed, and both exhausts led to the other condenser only, thus enabling repairs to be made without stopping any engine. The shut-off valves are shown in Figs. 6 and 7.

The diameters of cylinders of the reciprocating engines in the *Otaki* are 24½ inches, 39 inches and 58 inches, respectively, with a stroke of 39 inches. The cylinders of the *Orari* and *Opawa* are 24½ inches, 41¼ inches and 69 inches in diameter, respectively, and the stroke 4 feet. The ratio of high-pressure to low-pressure cylinders is thus 1 : 5.6 in the *Otaki* and 1 : 7.93 in the sister ships fitted with reciprocating engines.

The turbine fitted in the *Otaki* is, generally speaking, of the ordinary Parsons low-pressure type. The diameter of rotor drum is 7 feet 6 inches, and the length of blades 4¾ inches in the first expansion of 12 11/16 inches in the last expansion.

A novel feature is that the drum is completely closed at both ends, and any leakage past the dummy is led away by an external pipe to the condenser. The object of this is to reduce the loss due to the cooling effect of the condenser on the internal surface of the drum, and to prevent corrosion.

The turbine spindles are packed with soft packing, separated into two parts by a metallic lantern ring. This ring enables the shaft to be surrounded by water between the two divisions of the packing, thus forming a water seal. The water is supplied under a slight head, and a gauge at each stuffing box indicates the head of water and whether air leakage is occurring. A full-sized model of the arrangement was kept running for a month on shore before it was decided to adopt this method of packing, which has been found to work very satisfactorily on service. The stuffing box is shown in Figs. 8 and 9.

The lubrication of the turbine bearings is on the adjustable sight-feed system, and the oil pump is worked off the turbine

shaft. An alternative oil pump is worked from the air-pump lever of one of the reciprocating engines.

The condensers were made to the design of the Contraflo Condenser Company, and the total cooling surface of the two consenders is 6,000 square feet. Figs. 10, 11 and 12 show the design of the condensers.

There is one single-acting air pump of the ordinary bucket type worked from each set of main engines, the diameter of the bucket being 26 inches, and the stroke 19½ inches.

The circulating pumps have 16-inch suctions and discharges, and the diameter of the impellers is 48 inches.

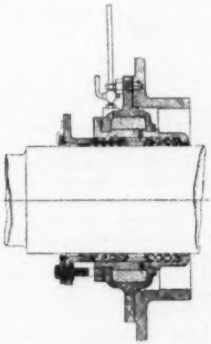


Fig. 8.—Metallic Packing of Turbine Spindles.

The feed pumps are of Woodeson's patent type.

A surface-feed heater is fitted in connection with the exhaust from auxiliary engines.

Tables I and II give some particulars of the results of the trials.

The *Orari* obtained a mean speed of 14.6 knots on the measured mile at Skelmorlie; while the *Otaki*, under the same conditions, with apparently greater ease, obtained a mean speed of over 15 knots for a total water consumption per hour of 6 per cent. less than the *Orari;* while the total water consumption per hour in the *Otaki*, at 14.6 knots, was 17 per cent. less than in the *Orari* at the same speed. The total water con-

sumption referred to here and in subsequent comparisons is that for the main and auxiliary engines.

In the *Otaki* the total water used in all the engines was measured by tanks during the trials, and the number of strokes of the feed pumps was also observed.

Table I.—Particulars of Progressive Runs on the Measured Mile at Skelmorlie, 31st October, 1908.

.....	Mean of 2 runs.	Mean of 2 runs.	Mean of 2 runs.	Mean of 2 runs.
Total horsepower (I.H.P. of reciprocating engines plus S.H.P. of center shaft)	6,857	5,348	4,704	3,282
Speed, in knots	15.02	14.278	13.829	12.518
Revolutions, port	103.0	96.2	93.1	84.6
starboard	103.5	97.9	93.5	83.4
center	224.5	209.7	197.2	172.1
Total water consumption per hour, by tanks (taken from curve), pounds	82,000	67,300	60,200	44,600
Total water consumption per hour, from strokes of feed pumps (taken from curve), pounds	84,700	69,300	62,300	48,200
Water per H.P. per hour, by tanks, pounds	11.95	12.6	12.8	13.6
pumps, pounds	12.35	12.95	13.25	14.7
Mean absolute pressure in H.P. receivers, by indicator diagrams, pounds	193.0	178.0	166.0	135.0
Absolute pressure at turbine inlet, measured by column of mercury, pounds	9.5	7.62	6.76	5.0
Vacuum at exhaust end of turbine, measured by column of mercury, inches	28.1	28.2	28.4	28.5
Vacuum in condenser, by gauge, inches	28.2	28.4	28.3	28.5
Barometer, inches	30.06	30.06	30.06	30.06
Temperature of sea water, degrees Fahrenheit	56.0	56.0	56.0	56.0
circulating disch., deg. F	70.0	67.0	70.0	70.0
feed water at hotwell, deg. F	72.0	70.0	73.0	74.0
Draught of ship, forward, feet and inches	...	18-10	...	...
aft, feet and inches	...	21-04	...	...
mean, feet and inches	...	20-01	...	...
Displacement, tons	...	11,716	...	...
Water consumption per I.H.P. per hour, based on I.H.P. of *Orari:*				
By tanks, pounds	13.66	13.7	13.8	14.07
pumps, pounds	14.12	14.1	14.3	15.2

The water consumption as calculated from strokes of the pumps is in all cases greater than that obtained by tank measurements, the difference being about 3 per cent. at the higher speeds.

The water consumption was not measured by tanks in the

Orari, being estimated only from the number of strokes of the feed pumps. In order to make a fair comparison of the consumption in the two ships, it is necessary to consider the water as estimated from strokes of the feed pumps in both cases, or to correct the estimated consumption in the *Orari* to make it comparable with the tank measurements of the *Otaki.* In view

Table II.—PARTICULARS OF VARIOUS TRIALS.

.....	1	2	3	4	5	6
Total horsepower.....	6,857	7,054	6,030	5,420	4,100	3,460
Speed, in knots.....	15.02	15.09	14.71	14.25	13.35	11.7
Revolutions, port.....	103.0	104.9	99.7	96.7	90.1	82.9
starboard.....	103.5	104.5	101.1	97.5	89.6	83.5
center.....	224.5	228.0	214.0	203.1	179.0	169.4
Total water consumption per hour, by tanks (taken from curve), lbs..	82,000	83,900	75,200	66,700	...	...
Ditto from strokes of pumps (taken from curve), pounds.....	84,700	86,700	77,600	68,700	56,400	49,800*
Water per H.P. per hour :						
By tanks, pounds.....	11.95	11.9	12.47	12.3	...	...
pumps, pounds.....	12.35	12.3	12.87	12.67	13.75	14.4*
Coal consumption per hour, lbs.....	...	...	...	...	...	4,800
Coal consumption per H.P. per hour, pounds.....	...	...	...	...	...	1.387
Mean absolute press. in H.P. receivers, by indicator diagrams, lbs.....	193.0	196.0	189.0	195.0	...	175.0
Absolute pressure at turbine inlet, measured by mercury col., lbs.....	9.5	9.8	8.5	7.5	5.75	5.2 (by gauge.)
Vacuum at exhaust end of turbine, measured by col. of mercury, ins..	28.1	28.0	28.1	28.2	...	...
Vac. in condenser, by gauge, ins....	28.2	28.2	28.35	28.4	28.75	28.1
Barometer, inches.....	30.06	30.0	30.0	30.1	30.2	...
Temperature of sea water, deg. F...	56.0	56.0	56.0	56.0	56.0	55.0
Temperature of circ. disch., deg. F..	70.0	74.0	70.0	70.0	70.0	65.0
Temperature of feed water at hot-well, degrees Fahrenheit.....	72.0	74.0	73.0	73.0	73.0	70.0
Draught of ship, for'd, ft. and ins...	18–10	18–10	...	...	...	22–06
aft, feet and ins....	21–04	21–03	...	...	...	23–10
mean, ft. and ins..	20–01	20–00	...	...	...	23–02
Displacement, tons.....	11,716	11,690	...	...	...	13,710
Water per I.H.P. per hour, based on I.H.P. of *Orari :*						
By tanks, pounds.....	13.66	13.8	13.65	13.7	...	...
pumps, pounds.....	14.12	14.25	14.1	14.1	14.54	...

* These are actual results.

Column (1) full speed on mile, mean of two runs, Oct. 31, 1908.
(2) full speed on mile, mean of two runs, Nov. 2, 1908.
(3) mean of two runs on mile, Nov. 2, 1908.
(4) continuous run of one hour, Nov. 2, 1908.
(5) continuous run for two hours, Nov. 5, 1908.
(6) run from Clyde to Liverpool, 14 hours, Nov. 21, 22, 1908.

of the greater accuracy of tank measurements the latter method of comparison has been adopted.

The corrected total water consumption of the *Orari,* at a speed of 14.6 knots, was 88,300 pounds per hour, or 16.5 pounds per I.H.P. per hour. The latter figure is about the mean of the results obtained under similar conditions in a

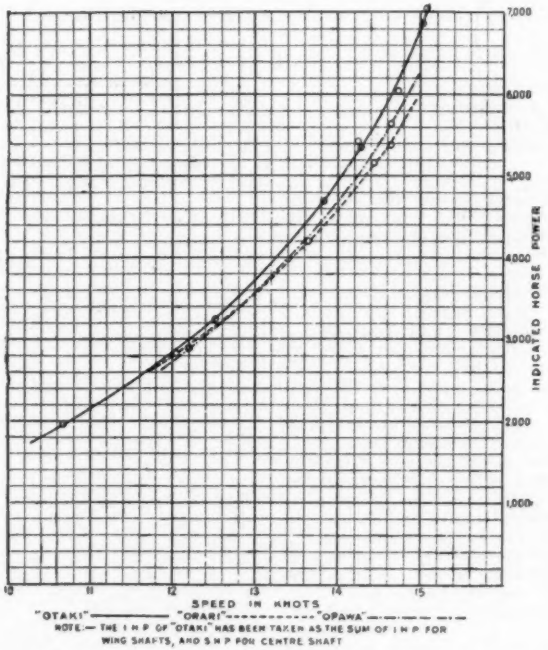


Fig. 14.—TOTAL WATER CONSUMPTION AND SPEED, S. S. "OTAKI."

number of vessels with machinery of similar design, and so may be regarded from a practical point of view as sufficiently reliable for purposes of comparison.

On the run from Clyde to Liverpool, with the vessel partly loaded, on November 21st and 22d, 1908, at about half the full power, records were taken for fourteen hours and the coal was measured.

The water was estimated from the total number of strokes of the feed pump during the period.

The evaporators were not in use.

The coal consumption works out at 1.387 pounds per H.P. per hour for all purposes, but, as will be seen from the particulars of water consumption, a lower coal consumption per H.P. might be anticipated at higher power.

The coal used was Scotch, of the following quality, as analyzed by two independent analysts: B. T. U. 13,446—13,617.

Accurate comparisons of coal and water consumptions per horsepower in two vessels driven by reciprocating engines and steam turbines respectively, are rendered difficult by the distinction between shaft horsepower and indicated horsepower. The actual shaft horsepower of the reciprocating engines is seldom known, and an assumed ratio of shaft horsepower to indicated horsepower, based on general information, may lead to error. Further, as the efficiencies of the propellers may differ sensibly, and we are concerned with the efficiencies of the whole propelling apparatus, the comparison of the two systems cannot be based on shaft horsepower alone.

If it be desired to compare the coal or water consumption of two vessels in relation to power, then it appears preferable to use the known indicated horsepower of the reciprocating-engined ship as a basis, provided the propellers are of normal design. We are thus eliminating any difference in efficiency of propellers, and dealing with power in the terms which custom has rendered familiar. In the case of two sister vessels the same horsepower would be used for both ships, and this method of comparison would therefore give the same results as comparison by total consumption.

From a design point of view, the most reliable comparison appears to be one based on water consumption per E.H.P. where this is available, while from a practical point of view it is obvious that in sister vessels, with similar boiler installations the coal consumption per mile or per hour at the same speed affords the best comparison of the two systems.

Fig. 15 shows the I.H.P. and speed curves of the *Otaki, Orari* and *Opawa.*

The proportion of total power developed in the turbine of the *Otaki* was found to vary with the speed. At full power this proportion was about one-third; while at very low speeds the turbine was doing only a small proportion of the work.

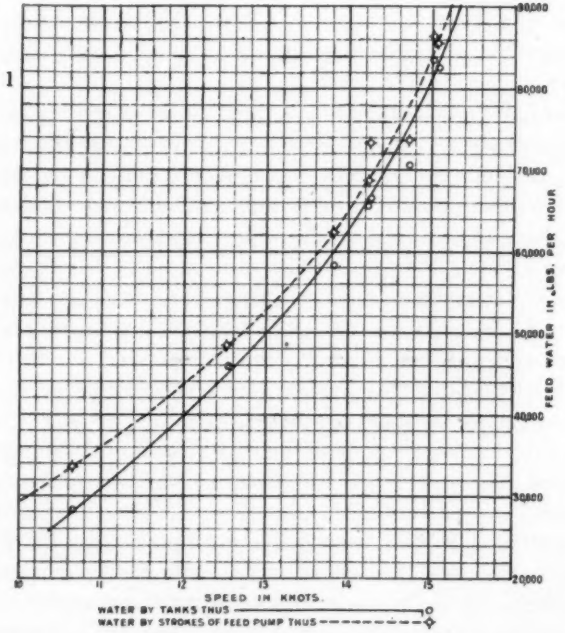


Fig. 15.—INDICATED HORSEPOWER AND SPEED, S. S. "OTAKI," "ORARI" AND "OPAWA."

At a speed of 14.6 knots, the indicated horsepower in the *Orari* was 5,350, and the corresponding power in the *Otaki* was 5,880. At this speed the E.H.P. was 3,210 in the *Orari* and 3,350 in the Otaki, the propulsive coefficients being thus 60 per cent. and 57 per cent. in the respective ships. The propulsive coefficient in the *Otaki* at full speed, fell to 54 per cent.

Applying now to the trial results of the *Otaki* and *Orari,*

at a speed of 14.6 knots, the various methods of comparison referred to:

(1) The comparison based on total water consumption and speed has already been dealt with.

(2) A water consumption of 21.9 pounds per E.H.P. per hour in the *Otaki* correspond to 27.5 pounds per E.H.P. per hour in the *Orari,* the gain on this basis being, therefore, 20 per cent.

(3) The lowest line of Tables I and II shows the water consumption per I.H.P. in the *Otaki,* taking the I.H.P. as that obtained in the *Orari* for corresponding speeds. On this basis a water consumption of 13.7 pounds per I.H.P. per hour in the *Otaki* corresponded to 16.5 pounds per I.H.P. per hour in the *Orari—i.e.,* the gain in the *Otaki* was 17 per cent.

A summary of the results of these comparisons is given in Table III.

Table III.—Summary of Comparisons of the "Otaki" and "Orari" for a Speed of 14.6 Knots on the Measured Mile.

...	E.H.P.	I.H.P.	Propulsive coefficient.	Water consumption.		
				Total, per hour.	Per E.H.P., per hour.	Per I.H.P. of *Orari*, per hour.
			Per cent.	*Lbs.*	*Lbs.*	*Lbs.*
Otaki.	3,350	5,880	57	73,300	21.9	13.7
Orari.	3,210	5,360	60	88,300	27.5	16.5
Gain per cent. in *Otaki*.....				17	20.0	17.0

Note.—Columns marked * do not take into account the difference of E.H.P. in the two ships; these two methods of comparison should show the same gain.

Table IV shows the initial pressures, range of temperature, etc., in the different cylinders as taken from two sets of indicator diagrams at the full speed in the *Otaki.* It will be observed that the range of temperature in the low-pressure cylinders of the reciprocating engines is about the same as in the other cylinders; also that the theoretical heat available in the turbine is over one-third of the total available heat.

Table IV.—Initial Pressures and Range of Temperature in Cylinders of the "Otaki" at Full Speed.

.....	Reciprocating engines.				Turbine.	
	H.P., initial press. (Diagram.)	M.P., initial press. (Diagram.)	L.P., initial press. (Diagram.)	L.P., exhaust press. (Diagram.)	Initial pressure. (Mercury col.)	Exhaust pressure. (Mercury col.)
Absolute pressures, pounds.....	196	91.5	36	11.5	9.8	1.0
Corresponding temperatures assuming saturated steam. Degrees Fahrenheit (approximately).....	380	321.0	261	200.0	192.0	102.0
Temperature range, degrees.....		59	60	61	8 (drop)	90
Theoretical available heat units per pound of steam, B.T.U. (approximately).....			200			120

Turning now to the performance of the *Otaki* on service, the coal consumption on the voyage from Liverpool to Teneriffe was 11 per cent. less than the mean for the sister vessels *Orari* and *Opawa* under similar conditions and at practically the same speed.

On the whole round voyage from Liverpool to New Zealand and back, the average speed maintained by the *Otaki* was 11.09 knots. This speed does not represent the sea speed which the vessel is capable of maintaining, the power required being only about half that developed on trial, nor is it the most suitable for obtaining the lowest consumption of coal per H.P., but it was the speed arranged to suit the owners' conditions of service. The initial pressure in the turbine was only about 4.5 pounds absolute.

A careful comparison of the coal consumption for the round voyage with that of the sister ships on similar voyages at the same speed shows an apparent gain of about 8 per cent. This gain represents about 500 tons of coal for the round voyage,

and as the ship leaves England with sufficient coal for the outward run, would permit of the cargo carried being increased one-half that amount. Compared, however, with the results on trial and on the voyage as far as Teneriffe, this result appears disappointing. It is, however, not really so, as there were various adverse circumstances to account for the apparent falling off in economy which it is not proposed to enter into here, as they were not attributable to the type of machinery. One contributing cause, however, was that, on docking, the vessel was found to have an abnormally foul bottom.

It is confidently anticipated that a greater gain in economy will be obtained in future.

The engines made a non-stop run outwards from Teneriffe to New Zealand, a distance of 11,669 miles as logged, which is probably the longest continuous run yet made by a marine turbine. The turbine worked perfectly satisfactorily throughout the whole round voyage.

The condensers were designed with a view to maintaining a high vacuum in tropical waters, and the following are typical records of the results obtained with sea water of various temperatures:

Height of barometer, ins....	30.02	30.05	30.09	30.00	30.09	29.85	29.65
Vacuum, inches.....	28¼	28¼	28¼	29.0	29.0	29.0	29.0
Temperature of water in hotwell, degrees Fahr.....	103.0	98.0	94.0	86.0	83.0	74.0	71.0
Temperature of sea water, degrees Fahrenheit.....	84.0	80.0	76.0	68.0	63.0	56.0	53.0
Temperature of circulating discharge, deg. Fahr	98.0	94.0	91.0	82.0	80.0	68.0	69.0
Vacuum, corrected to 30-in. barometer and reduced .35 inch to allow for possible error in gauges.....	27.88	28.1	28.3	28.65	28.56	28.8	29.0

The revolutions of the main reciprocating engines, when these records were taken, were about 81 per minute, and the revolutions of the circulating pump 150 per minute.

The vacuum is given as recorded, but a slight reduction should be made in each case for errors in gauges. The *maximum* error observed in the gauges on trial as compared with

the mercurial column was .35 inch, while the mean error was under .2 inch.

From these records, even with a deduction corresponding to the *maximum* error observed, it appears that practically the possible vacuum with the respective hot-well temperatures was obtained. The drains from the turbine are always open to the air-pump suction, so that the temperature of the water in the hot well was probably slightly higher than that of the water at the bottom of the condenser. Further, there may be slight errors due to the vacuum being recorded to the nearest quarter of an inch.

The vacuum gauges are placed at about the level of the middle of the condenser.

It will be admitted that these results are very satisfactory, but it has to be remembered that the vessel was only steaming at about half power.

Figures for vacuum obtained on the trials are given in Tables I and II.

As regards the feed heater, with a pressure of from 8 to 12 pounds in the auxiliary exhaust, the temperature of the feed water was raised 210 to 215 degrees Fahrenheit, while using only part of the auxiliary exhaust steam. Some gain in economy would be obtained by leading the surplus auxiliary exhaust steam to the turbine.

The dimensions of the engine room in the *Otaki* were the same as in her sister ships, and were found sufficient to admit of an arrangement of machinery which gave satisfactory access to all working parts.

The total weight of machinery in the *Otaki* was about 30 tons more than in her sister ships—an increase of about 3.25 per cent. In view of the greater economy of the combination system the boilers of the *Otaki* might have been reduced. Allowing for this, the total weight of an installation of the combination system would not exceed that for reciprocating engines. The saving in bunker capacity would more than balance the loss in cargo-carrying capacity due to the three tunnels, and hence in future vessels of similar design no increase in

length would be necessary. In the *Otaki,* however, the owners desired no alteration in the bunker capacity or the boilers—hence the increased length.

The possible difficulty in maneuvering has been advanced as an objection to the combination system. No difficulty of any kind in connection with maneuvering was experienced in the *Otaki.* The maximum power cannot be developed when going astern, but it was found that astern power sufficient for all practical purposes had been provided.

Another objection which has been raised is the possible effect of oil from the reciprocating engines on the turbine blades, and possible clogging of the first row of fixed blades by oil and other foreign matter. Pending experience on this point it was decided not to fit oil separators or strainers in the *Otaki,* a large pocket only, with a baffle and examination and cleaning doors, being provided at the lower end of the steam-inlet pipes to the turbine. After the trials it was found that there was no collection of oil or other foreign matter. A further examination was made at Teneriffe and another on arrival in New Zealand with similar results. On arrival in England the turbine cover and rotor were lifted and a thorough examination made. There was a slight amount of deposit in the inlet-pipe pockets, but the blades were in good condition and quite clean, with the exception of a few in the first rows at the lowest part of the turbine on which there was a very thin layer of an oily deposit. From the experience on this voyage there appears no reason to anticipate any trouble from this cause.

The results, on the whole, taken in conjunction with the performance of the vessel on service, are considered sufficient to show that a high degree of economy may be anticipated in the combination type of engine—an economy probably higher than has been obtained in any other type of marine engine for vessels of this class. There appears little doubt that in vessels of low speeds, for which Parsons turbines alone are unsuitable, and where economy is of primary importance, this system can be advantageously employed.

STEAM YACHT "WINCHESTER."

The Fast Steam Yacht *Winchester,* constructed for an American owner by Messrs. Yarrow, of Glasgow, passed her trials last week in the Firth of the Clyde, and will leave Glasgow shortly for New York. She will be towed across the Atlantic.

She is built on torpedo-boat lines, and her dimensions are as follows: Length, 165 feet; beam, 15 feet 6 inches.

Her estimated speed was 26 knots, and the mean speed obtained on a two-hours trial was 26.7 knots.

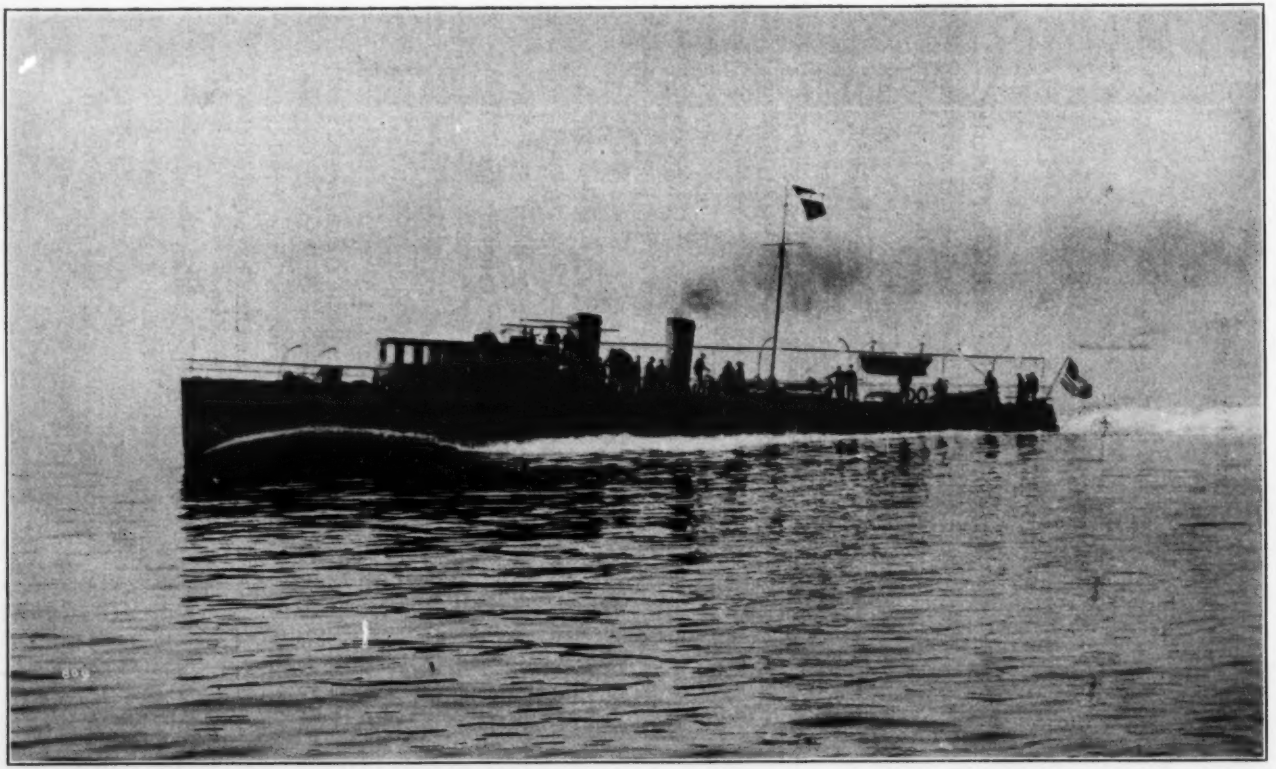
The propelling machinery consists of three turbines of Parsons type (constructed by Messrs. Yarrow), with three shafts, one propeller on each shaft. The port shaft is actuated by a high-pressure turbine, the center shaft by a low-pressure turbine in which is incorporated an astern turbine, and the starboard shaft by a middle-pressure turbine.

Steam to the turbines is supplied by two Yarrow water-tube boilers, constructed for burning oil-fuel exclusively.

The accommodation for the owner, which is below deck abaft the machinery space and is very commodious and handsomely decorated, consists of a double stateroom, two single staterooms, drawing room, bathroom and toilet rooms. There is a teak deck house forward, 25 feet long, containing dining room, captain's cabin and pantry. The quarters for the officers and crew, and the galley, are below the main deck, forward.

The vessel has a compiete electric-light installation and is heated by steam.

It is greatly feared that there will not be many yachts built in this country for America in the future, on account of the excessive duty which has been placed on yachts built out of America.



STEAM YACHT "WINCHESTER."

MERCHANT SHIPS.

The Orient Company's Australian Mail Liner Osterley.—The third of the five vessels ordered by the Orient Steam Navigation Company, Limited, for the new fleet which is to conduct the new mail service from Australia, under contract with the Commonwealth Government, closely resembles the *Orsova* and the *Otway*—the two preceding ships built on the Clyde for the service—having a length over all of 552 feet, a breadth of 63 feet 3 inches, and a depth to the shelter-deck of 46 feet.

The Propelling Machinery.—The propelling machinery of the vessel is a notable feature. The engines, of which there are two sets, are quadruple expansion, and of the boilers, four are double ended and two are single ended.

The four cylinders are arranged according to the Yarrow-Schlick-Tweedy system, and piston valves are fitted to the high-pressure and first-intermediate cylinders, and slide valves to the others. The dimensions of the cylinders are: High-pressure, 28¾ inches in diameter; first intermediate pressure, 41 inches in diameter; second intermediate pressure, 58½ inches in diameter; low pressure, 84 inches in diameter; all having a stroke of 60 inches. All the cylinders have separate liners and are steam jacketed. All the working parts are of forged steel; the crank shaft, as shown in Fig. 1, is built up in two sections, the diameter of the shaft being 17½ inches. The thrust blocks are of the horse-shoe type. The propellers have cast-steel boss and three magnanese-bronze blades.

In addition to the usual lubricating fittings, the engines are provided with a set of "Axiom" lubricating gear having connections to the main bearings, crank-pins, crosshead and guides. All the rods in the main engines and pumps are fitted with United States duplex packing.

The main condensers, as shown, are independent, of cylindrical section, and are made of steel boiler plate, with cast-iron water ends. They are carried on brackets bolted to the back columns. The circulating pumps, by Messrs. W. H. Allen, Son & Co., Limited, are of large size, each being sufficient to supply all the circulating water for both sets of engines when they are running at their normal power. The pumps are connected so that either may be used on both main condensers.

None of the pumps are worked off the main engines—all are independent. There are one pair of Weir's air pumps, for each set of main engines, two hotwell pumps, three main feed pumps, and one direct-contact heater—all supplied by Messrs. G. & J. Weir, Limited. The feed-filters, made by Messrs. Carruthers & Co., are fitted in the feed-suction system. The hotwell pumps draw from the main and reserve-feed tanks and discharge to the heater only. The main-feed pumps draw from the heater, hotwell, or reserve tanks, and discharge direct to the boilers; one of these pumps is also arranged to discharge through the feed heater if required. Large air vessels are fitted to all the pumps. A Carruthers duplex pump is provided for supplying the boilers in port. The bilge pumps include one set of three-throw single-acting plunger pumps, driven by a three-throw crank shaft, to which an electric motor is connected by worm gearing, and a duplex pump of the ordinary type. These pumps were supplied by Messrs. J. H. Carruthers & Co.

The deck machinery, refrigerating engines and electric-light and power engines are provided with a separate system of steam and exhaust pipes, including an auxiliary surface condenser, with air pump and circulating pumps complete. The exhaust system is arranged so that the waste steam may be directed to the main condenser, the low-pressure valve casings, the feed heater, or to the atmosphere. The dynamos, of which there are four, were supplied by Messrs. W. H. Allen, Son & Co., Limited. They have a combined output of about 300 kilowatts. Weir's evaporators, two in number, each of 40

tons capacity, and a fresh-water distiller working in conjunction with the evaporators, are provided for supplying extra feed water for the boilers and fresh water for use in the ship.

The Sanitary Machinery.—The sanitary pumps are two in number, of Drysdale's centrifugal type, and are electrically driven. The general and ballast pumps are each capable of discharging 200 tons per hour; they, together with the fire pump and two fresh-water pumps, are all of Carruthers' duplex type.

In connection with the sanitary arrangements an interesting detail is fitted in this and the other ships in the form of a patent automatic turbine water-closet flusher. It is manufactured by Messrs. Steven & Struthers, Kelvinhaugh, Glasgow, and is applicable alike to steamships and works, and especially where seamen, stokers and workers are generally careless in applying ordinary flushing arrangements.

The apparatus consists of a circular casing having ten or twenty outlets, each outlet being connected to a closet or soil pipe by a pipe 1½ inches in diameter. Inside of the casing is a controlling cylinder revolving on ball bearings, and having a single port, so that every time this port comes opposite an outlet the flush takes place. The water turbine is placed in the connection to the supply pipe from the main tank. The system of driving the cylinder is ingenious. The water flowing through the turbine causes it to revolve, but as its speed would be too great for driving the controlling cylinder direct, the connection is by means of its spindle acting through a worm and worm-wheel reducing gear, which gives the desired speed of rotation. To ensure that each pipe connection has a sufficient duration of flush, the worm wheel is attached to a ratchet, and this gives an intermittent action to the controlling cylinder and sufficient length of time for flushing each pipe. The apparatus is very simple, and has no stuffing boxes, an ingenious arrangement carrying away all leakage to one of the soil pipes. In the event of the supply tank not being high enough to give head for driving the turbine, provision is made so that a small electric motor can be attached.

Table I.—Progressive-Speed Trials on Measured Mile at Skelmorlie, June 18, 1909.

No. of run and direction.	1st run —Up.		2d run —Down.		3d run —Up.		4th run —Down.		5th run —Up.		6th run —Down.		7th run —Up.		8th run —Down.		9th run —Up.		10th run —Down.		11th run —Up.		12th run —Down.	
Port and starboard engines	P.	S.	P.	S.	P.	S.	P.	S.	P.	S.	P.	S.	P.	S.	P.	S.	P.	S.	P.	S.	P.	S.	P.	S.
Steam pressure at boilers, lbs.	200	205	205	195	200	205	200	205	200	205	200	205	190	180	200	205	215	205	215	215	215	215	215	215
Vacuum, ins.....	27	24	27	25	27	25	26½	25	27½	27	28	28	28	28	28	28	28	27½	28	28	26½	27	27	26½
Revolutions.....	60.2	61.0	61.0	62.2	72.0	70.6	69.5	70.0	77.0	77.0	78.0	78.0	82.8	83.6	83.0	84.0	88.0	88.8	87.4	88.4	91.8	94.2	93.0	94.8
Total I.H.P.....	1744	2040	1745	1898	2952	2640	2752	2687	3652	3555	3832	3651	4601	4580	4708	4917	5697	5350	5577	5691	6866	6746	7027	6942
Collective I.H.P.	3,784		3,643		5,592		5,439		7,207		7,483		9,181		9,625		11,047		11,268		13,612		13,969	
Mean I.H.P. (two runs).	3,713				5,515.5				7,345				9,403				11,157				13,790			
Mean revolutions per min.	61.1				70.52				77.5				83.3				88.15				93.5			
Speed of ship, knots.....	12.99	13.04			15.12	14.81			16.0	16.29			16.82	17.65			17.57	18.56			18.27	19.25		
Mean speed, knots.	13.01				14.96.				16.4				17.23				18.06				18.76			

Table II.—17¼-Knot Trial, June 19, 1909. Runs Between the Cumbrae and Cloch Lights (13.66 Nautical Miles).

Number and direction of run.	1st. Cumbrae to Cloch.		2d. Cloch to Cumbrae.		3d. Cumbrae to Cloch.		4th. Cloch to Cumbrae.		Trip round Ailsa Craig. Four hours' run.							
									1st hour.		2d hour.		3d hour.		4th hour.	
Port and starboard engines	P.	S.	P.	S.	P.	S.	P.	S.	P.	S.	P.	S.	P.	S.	P.	S.
Steam pressure at boilers, lbs.	215	215	215	215	212	215	210	215	215	215	215	215	215	215	215	215
Vacuum, inches	26.6	26.75	27.0	27.0	27.0	26.75	27.0	27.0	26.0	25.5	25.5	26.0	25.0	26.25	26.0	26.25
Revolutions	88.6	90.5	88.8	90.3	88.8	90.7	88.6	90.2	89.39	90.8	89.3	90.3	89.5	91.2	89.2	90.9
Total I.H.P.	6,123	6,105	6,136	6,051	6,220	6,068	6,144	6,114	6,283	6,098	6,194	6,126	6,173	6,192	6,263	6,171
Collective I.H.P.	12,228		12,187		12,288		12,258		12,381		12,320		12,365		12,434	
Mean coll. I.H.P.	12,207				12,273				12,350.5				12,399.5			
	12,240								12,375							
Speed of ship, kts.	18.3		18.2		18.1		18.50									
	18.25 mean.				18.32 mean.											

The Boilers.—There are six boilers, two of which are single-ended and four double-ended. Each boiler is 16 feet 6 inches in diameter. The single-ended boilers are 11 feet 6 inches long, the double-ended are 20 feet 2 inches long. There are forty furnaces, giving, on 5-foot 6-inch bars, a grate area of 680 square feet, while the total heating surface is 31,000 square feet. The boilers are fitted with Howden's system of forced draft. They are five fans, made by Messrs. Howden, and each is driven by a separate motor.

Steam Trials.—The steam trials were carried out with promptitude and with uniform success. On the 18th inst. the vessel carried out a series of progressive speed trials at about 13, 15, 16, 17 and 18 knots, and at full speed. The results of these trials, which are very favorable, are set out in Table I. It will be noted that at full speed the vessel, developing 13,790 horsepower at 93.5 revolutions, attained a mean of 18.76 knots on the two mile runs, which shows high efficiency. The result anticipated by the contract was 17½ knots. On the following day the vessel had to continue for 10 hours at approximately this speed, and during this period four runs were made on the 13.66-nautical-mile course between the Cloch and the Cumbrae Lights. On this trial the mean power maintained was 12,240 H.P., and the mean speed 18.28 knots, as shown in Table II. It should be noted that these trials were carried out at the mean service draught and displacement.

Following upon the Cloch and Cumbrae trials, various trips were made further down the Firth of Clyde around Ailsa Craig, during which the power of the preceding trial was more than maintained, as shown in Table II. The vessel left the Clyde on the conclusion of these trials, and on the way round to London carried out the 24-hours' coal-consumption trial at service speed and at service draught. A requirement of the contract was that at 16½ knots speed the coal consumption should not exceed 1.4 pounds per indicated horsepower, and this condition was fulfilled as in the preceding ships. The results are tabulated:

Mean draught (as on service), 24 feet 3 inches; mean dis-

placement, 15,300 tons; mean indicated horsepower, 8,640 I.H.P.; speed (according to revolutions), 16¼ knots; coal consumption per I.H.P. per hour, 1.134 pounds; distance steamed in 24 hours, 390 miles; coal consumed in 24 hours, 105 tons.—"Engineering."

OBITUARY.

REAR ADMIRAL ROBERT WILEY MILLIGAN, U. S. N., RETIRED.*

Rear Admiral Robert W. Milligan, who died at his home in Annapolis, Md., on October 14th, 1909, was one of the well known and best beloved members of the Society, whose loss is keenly felt by all who knew him. His death was due to an acute attack of heart trouble from which he had been a great sufferer for several years. He had been confined to his room for nearly a week, but there was no uneasiness felt in regard to him, therefore death came as a great shock to his family and intimate friends. The funeral was held on October 17, 1909, with military honors. Interment was in the Naval Academy cemetery. The pallbearers were Rear Admiral Harry Knox, U. S. N., Rear Admiral Perry Garst, U. S. N., Colonel McLane Tilton, U. S. M. C., Captain Walter F. Worthington, U. S. N., Commander Frank W. Bartlett, U. S. N., Surgeon Andrew R. Wentworth, U. S. N., Lieutenant Commander Cleland N. Offley, U. S. N., and Lieutenant Henry N. Jenson, U. S. N.; the two last named having served with him on the *Oregon* during the Spanish War.

Admiral Milligan was born in Philadelphia in 1843, entering the Navy in 1863, as Third Assistant Engineer. During the Civil War he served on the U. S. S. *Mackinaw*, and participated in the attack on Fort Fisher, the fall of Wilmington and the fall of Richmond and Petersburg.

The outbreak of the Spanish War found him serving as Chief Engineer of the *Oregon*, on the Pacific Coast, and he continued on this vessel throughout the war. The remarkably successful voyage from Puget Sound to join the fleet in the operations in Cuban waters and the splendid record made by

* Written by Lieutenant Commander C. N. OFFLEY, U. S. N.

the ship on the blockade off Santiago, and in the battle of July 3d, 1898, were due largely to his perseverance, engineering skill and ability, as well as to his great experience and sound professional judgment. After the close of the Spanish American war, he was advanced five numbers in his grade for eminent and conspicuous conduct in battle. He then performed various duties until his retirement in 1905 with the rank of Rear Admiral.

He married a sister of Charles A. Dubois, formerly mayor of Annapolis, who survives him. He is also survived by two daughters, Mrs. Tench Tilghman, of Norfolk, Va., and Miss Mollie Milligan, of Annapolis.

Those who knew him, and especially those who had the good fortune to serve with him, will always remember his lovable qualities, his consideration for and personal interest in his subordinates, his high moral character, combined with strict and careful attention to the details of his profession, making it both a pleasure and an education to work with him.

Many incidents could be told of his faithful and efficient performance of duty and of the high esteem and admiration with which he was regarded by all associated with him, but the condition in which the *Oregon* reached the blockading fleet and her subsequent performance in battle, as regards machinery, is the highest tribute that can possibly be paid to the signal ability of Chief Engineer Milligan.

The following incident will show the high professional judgment of the man, and of his intuitive knowledge of what was required of a blockading ship. One of the boilers had started a leak in the tube ends and back connection. As usual, the defects were repaired immediately. Upon the completion of the work fires were at once started and this boiler connected with the others as soon as the proper pressure was reached. Succeeding events sustained the judgment of the Chief Engineer, since the exceptional speed made by the *Oregon* during the chase of the Spanish squadron was only possible because the vessel had full steam on all boilers.

The distinguished service rendered to the Navy and to the

Nation by Rear Admiral Milligan on board the *Oregon* during the Spanish-American war is held in the highest appreciation by the Society, and the professional esteem, military regard and personal affection in which he was held by the officers and men of the engine department who had the honor of serving under him before and during the battle of Santiago is a high tribute to his character.

COMMANDER FREDERIC CHARLES BIEG, U. S. NAVY.

Commander Frederic Charles Bieg, U. S. Navy, died at the Naval Medical School Hospital, Washington, D. C., on October 14, 1909.

Commander Bieg was born at St. Louis, Missouri, April 27, 1856. His parents were Valentine Nicholas Bieg and his wife Antonie, who came to the United States from Austria. He was a nephew of Lieutenant Colonel Adolph Ipold of the Austrian Army, and a cousin german of Dr. Franz Klein, late Austrian Minister of Justice.

He is survived by a wife and one son, Valentine N. Bieg, who is now a Midshipman in the U. S. Navy.

Commander Bieg entered the Navy as a Cadet Engineer, being appointed to the Naval Academy October 1, 1874. He was promoted to Assistant Engineer June 20, 1880; to Past Assistant Engineer October 21, 1890; and to Chief Engineer October 11, 1898. This last rank was changed to Lieutenant Commander March 3, 1899, by the Act of Congress approved on that date. He was promoted to Commander May 13, 1906.

He served through the Spanish-American War, and was, from September 27, 1904, to March 10, 1906, successively Chief Engineer of the Atlantic Training Squadron and the North Atlantic Fleet. His last duty was in the Bureau of Steam Engineering, Navy Department.

Commander Bieg was one of the few remaining members of the original Engineer Corps, and did much excellent work in the building up of the new Navy. He was among the first to advocate the establishment of an advanced course in eng-

ineering for young engineer officers. This plan has taken final shape in the new class recently ordered to the Naval Academy for advanced engineering study.

He was an accomplished linguist. The work on Marine Boilers of which he was the author is one of the standard books issued to the Naval Academy and to naval vessels.

Commander Bieg was an active member of the AMERICAN SOCIETY OF NAVAL ENGINEERS, where he served both as Secretary-Treasurer and as member of the Council.

His death is a distinct loss to the Naval Service.

LIEUTENANT CARL A. RICHTER, U. S. N.

The death of Lieutenant Carl A. Richter, U. S. N., member of the Society, occurred on board the U. S. S. *Marietta* on October 13, 1909, on which vessel he was serving as senior engineer officer.

Lieutenant Richter was an officer of excellent engineering ability and attainments and evinced a special interest in naval engineering matters. By his death the Society loses a valuable and interested member and the Naval Service an excellent and efficient officer.

BOOKS RECEIVED.

SCHIFFSTURBINEN.—BAUER AND LASCHE. (In German.) Published by R. OLDENBOURG, Munich and Berlin.

This valuable contribution to Marine Turbine literature by Dr. G. Bauer, Director of the Vulcan Works, and O. Lasche, Director of the turbine works of the A. E. G. Co., assisted by E. Ludwig and H. Vogel, gives many valuable details, methods of design, calculations and descriptions of the latest development of turbine application to marine service. The work is well illustrated with views of different turbines, their arrangement in vessels and also the details of construction, including shafting, propellers, condensers, valves, etc. The application of the A. E. G. (Curtis) turbine to marine purposes is extensively treated of. Applications of the turbine to marine auxiliaries such as air pumps, circulating pumps, feed pumps and blower engines are also described and discussed.

The book should be of special value as giving the very latest developments of marine turbine machinery as now developed in Germany. The book is in the nature of an appendix to Bauer's "Berechnung und Konstruktion der Schiffsmaschinen und Kessel." The book contains 200 pages and 104 illustrations.

ASSOCIATION NOTES.

At the meeting of the Society held on October 18, 1909, the result of votes on amendments to by-laws of the Society sent out for vote last May was announced as follows:

Change in paragraph 5, 146 in favor, 6 against.
Change in paragraph 6, 136 in favor, 19 against.

A motion was carried "Resolved, that it is the sense of this meeting that a dinner of the Society be given sometime during the next year."

The nominations for officers for year 1910, resulted as follows:

For President:

Rear Admiral H. I. CONE, U. S. N.

For Secretary-Treasurer:

Lieutenant H. C. DINGER, U. S. N.
Lieutenant ROBERT HENDERSON, U. S. N.

For Members of Council:

Commander R. S. GRIFFIN, U. S. N.
Commander W. S. SMITH, U. S. N.
Engineer-in-Chief CHAS. A. MCALLISTER, U. S. R. C. S.
Lieutenant Commander LLOYD CHANDLER, U. S. N.
Lieutenant Commander L. A. KAISER, U. S. N.
Lieutenant Commander U. T. HOLMES, U. S. N.
Lieutenant Commander J. K. ROBISON, U. S. N.
Lieutenant JOHN HALLIGAN, U. S. N.

The Annual Meeting of the Society for the counting of the votes will be held in the Bureau of Steam Engineering, at 4.30 P. M. Friday, December 31, 1909.

By order of the Council:

(Signed) J. K. BARTON, U. S. N.,
President.

H. C. DINGER, U. S. N.,
Secretary-Treasurer.

ANNOUNCEMENTS.

THE INTERNATIONAL CONGRESS OF MINING, METALLURGY, APPLIED MECHANICS AND PRACTICAL GEOLOGY, DÜSSELDORF, 1910.

The International Congress of Mining, Metallurgy, Applied Mechanics and Practical Geology, held at Liege in 1905, resolved, at the closing meeting on July 1st of that year, to accept the invitation of the Rhenish-Westphalia Mining Industry to hold the next Congress in Rhenish-Westphalia.

Acting upon this resolution, we have now the honor to inform you that the Congress will be convened at Düsseldorf, the last week in June, 1910, having been fixed as the date.

The Congress will be divided into the following Sections: (1) Mining, (2) Metallurgy, (3) Applied Mechanics, (4) Practical Geology.

The work of the Congress will be performed—

1. In general meetings, at which various papers of general interest will be presented.
2. In sectional meetings, for the purpose of discussing important problems relating to Mining, Metallurgy, Applied Mechanics and Practical Geology.
3. By making visits to scientific institutions and industrial undertakings, etc., and by excursions to districts of geological interest.

For the purpose of entertaining ladies accompanying members to the Congress a Ladies' Committee will be formed, which will endeavor in every way to render the stay of the lady visitors in Düsseldorf as agreeable and enjoyable as possible.

Any inquires for further information should be addressed to the "Arbeitsausschuss des Internationalen Kongresses, Düsseldorf, 1910, (Committee of Organization of the International Congress), Jacobistrasse 3/5, Düsseldorf, Germany."

ANNUAL MEETING OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

The thirtieth annual meeting of the Society will be held in the Engeneering Societies Building, 29 West Thirty-ninth Street, New York, December 7 to 10.

The entire social entertainment will be in charge of the members resident in and about New York, under the immediate direction of a local committee appointed by them, of which Mr. William D. Hoxie, is chairman. For Wednesday afternoon, December 8, an excursion is planned which members and guests will be asked to attend in a body, and during the balance of the time there will be opportunities for smaller parties to visit places of interest. In the evening there will be a lecture for members and guests upon the subject of Agricultural Machinery.

The professional papers assigned to the meeting are as follows: Tests on a Venturi Meter for Boiler Feed, Chas. M. Allen; The Pitot Tube as a Steam Meter, Geo. F. Gebhardt; Efficiency Tests of Steam Nozzles, F. H. Sibley and T. S. Kemble; An Electric Gas Meter, C. C. Thomas; Tan Bark as a Boiler Fuel, David M. Myers; Cooling Towers for Steam and Gas-Power Plants, J. R. Bibbins; Some Studies in Rolling-Mill Engines, W. P. Caine; An Experience with Leaky Vertical Fire-Tube Boilers, and the Best Form of Longitudinal Joint for Boilers, F. W. Dean; Testing Suction-Gas Producers with a Koerting Ejector, C. M. Garland and A. P. Kratz; Bituminous Gas Producer, J. R. Bibbins; The Bucyrus Locomotive Pile Driver, Walter Ferris; Line-Shaft Efficiency, Mechanical and Economic, Henry Hess; Pump Valves and Valve Areas and a Report on Cast-Iron Test Bars, A. F. Nagle.

In addition to the above papers there will be several valuable reports submitted by committees of the Gas-Power Section.